PREVENTIVE MAINTENANCE WORKSHOP

Environment
Ontario

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Minister

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INTRODUCTION

The Preventive Maintenance Workshop manual has been prepared as a home study and reference manual for plant operators and as the text for the related workshop.

The principal objective of the Workshop is to upgrade the knowledge of experienced water and wastewater treatment plant operators. The lesson objectives are clearly indicated at the beginning of each topic and tell the operator what he must know or do after having covered the topic. Upon completion of the Workshop, the trainee is expected to attain a minimum level of competence of 70% for the course.

This is a working course in which each person is expected to take an active part in subject discussions. It includes hands-on participation, in order to provide as much practical knowledge as possible of good maintenance procedures.

The Training and Certification Section wishes to acknowledge the assistance and contributions of the following in the preparation of the manual:

Ministry of the Environment:

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Various Manufacturers and Suppliers

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SUBJECT: **TOPIC: 1**

PREVENTIVE MAINTENANCE **TYPES OF PUMPS**

OBJECTIVES:

The trainee will be able to:

1. Recall the various types of centrifugal pumps and the principal application for each.
2. Identify the major components of a centrifugal pump and recall the function of each.
3. Recall the various types of positive displacement pumps and the most suitable application for each.
4. Identify the major components of various positive displacement pumps and the function of these components.

TYPES OF PUMPS

General

Basically pumps are categorized as either the centrifugal type or the positive displacement type (rotary or reciprocating). These types can be further classified according to the types of service for which they are designed or by features of their construction.

Pumps of the centrifugal type are:

1. Volute type centrifugal pumps

   These are the most common type in use today. An impeller discharges the fluid into a progressively expanding spiral casing. This is proportioned to reduce liquid velocity, changing it to pressure head in the volute. Subclassifications include open or closed impellers, single end-, or double-suction types, single or multi-stage. Single suction pumps have the liquid entering the impeller from one side. Double-suction pumps have liquid entering the impeller from both sides. In these cases, the impeller is built like two single suction impellers put back to back. Multi-stage pumps have two or more impellers on a common shaft, acting in series in a single casing. The liquid is conducted from the discharge of the preceding impeller or stage to the suction of the following stage, causing a pressure increase as it passes through each stage.

2. **Diffuser type centrifugal pumps**

These use stationary guide vanes surrounding the impeller. The gradually expanding passages change the direction of liquid flow, converting velocity energy to pressure head. Most common application today is in multi-stage, high pressure units. Efficiency is about the same as for volute-type. Both volute and diffuser types are relatively low capacity units, though a wide range of flows are available.

3. **Axial-flow (or propeller-type) centrifugal pumps**

These develop most of their head by the lifting action of the vanes. These low-head, high capacity units are essentially similar to Kaplan-type water turbines operating at high speeds. Usually of vertical design, they can be used in a variety of services, handling dirty liquids and those with solids contents.

4. **Mixed-flow type centrifugal pumps**

These are used mainly on large capacity, low-head installations. Usually built as vertical units, this type of pump uses both centrifugal force and the lift of the pump vanes to pump liquid. Impellers are single-end type, single or multi-stage. Either water or oil lubrication may be used on this type of unit, depending upon application.

5. **Turbine type centrifugal pumps**

These are higher capacity units with the impeller's vanes adding energy to the liquid. Use of this type of unit is largely restricted to clear liquids, with specific advantages where high-head operation is required. Both horizontal and vertical designs are available. Turbine pumps offer a cross between centrifugal and rotary design characteristics.

Rotary type positive displacement pumps are:

1. **Gear type rotary pumps**

These use two or more gears to provide positive-displacement, smooth flow movement of liquids. Operation is simple. Liquid fills spaces between gear teeth as they separate on suction side, is carried around and compressed as the teeth mesh. Gears may use spur, single or double-helical teeth depending upon design. Like all rotaries, is suitable for handling viscous fluids. Two, three and four lobe designs use synchronized turning of lobes to provide pumping action.

2. **Screw type rotary pumps**

These are available in single-screw designs (sometimes called progressive cavity pumps) as well as two and three-screw designs. This type of pump can handle very viscous, nonabrasive fluids, including such materials as cement grout. Close tolerances maintained on screw type pumps make handling of abrasive materials difficult.

3. **Vane type rotary pumps**

   These use vanes or buckets turned by a single rotor. Variations include swinging-vane (where hinged vanes swing out to trap liquid), sliding-vane where vanes are thrown against casing bore by centrifugal action, and shuttle-block pumps. Flexible vane pumps are also used for lower pressures than rigid-type vane pumps.

4. **Cam-and-piston type rotary pumps**

   These use an eccentric with a slotted arm at the top. Shaft rotation causes the eccentric to trap liquid in the casing, discharging it through the slot. This provides the steady discharge flow common to most rotary pumps. Other eccentric designs include use of eccentrics in flexible chamber which pumps by a squeezing action, and the flexible-tube pump where a flexible tube is squeezed by a compression ring on the eccentric.

Reciprocating type positive displacement pumps are:

1. **Direct-acting reciprocating pumps**

   These are frequently steam-driven units. Like all reciprocating pumps, they feature pulsating flow, relatively small capacity, and ability to handle high pressures. Increased head will decrease capacity of direct-acting units, unlike other types of reciprocating pumps. Direct-acting steam pumps have two sets of valves in the liquid end. Double-acting pumps are most common, with liquid being drawn in through suction valves as pistons move in, while liquid previously drawn in is discharged through the discharge valves.

2. **Power-type reciprocating pumps**

   These are driven from outside through a crankshaft or other device. Capacities range from low to medium flows, at medium to high pressure. These may be single acting or double-acting units. These units range from small adjustable-displacement types, for metering and proportioning to large units designed for marine boiler feed service. Output curve of power-pump designs approximates a sine wave, while direct acting types are relatively flat, with sharp drops at the end of piston stroke.

3. **Crank-and-flywheel type reciprocating pumps**

These are another major version of the power-type pump. The major difference is in driving mechanisms. The crank-and-flywheel is a specific type of drive, where a number of different mechanisms may be used for driving power-type pumps.

NOTE: **All reciprocating pumps can be single- or double-acting to smooth out pulsatings in flow. Duplex and triplex types are designed to smooth out flow still further. Special constant-delivery types are also available, which provide a flat output curve regardless of adjustment of stroke length.**

The various types of pumps in each category are discussed in detail in the following paragraphs.

CENTRIFUGAL PUMPS

General

Because of their adaptability to a wide variety of operations, low initial cost and easy maintenance, centrifugal pumps make up the most widely used type of pump in industrial plants, as well as municipal water and wastewater operations. In certain cases changing an installed centrifugal pump to handle heavier or more viscous materials, such as sludge or liquids containing granular material, may only require installation of an impeller of different size and shape.

Pump Components

Although types of centrifugal pumps differ in design, components are usually common. Referring to Figure 3-2 the parts are:

1. **Suction End (Pump Inlet)**

   This is the point at which the material pumped enters the pump. It is normally located near the centre of the casing. The diameter (or size) of this opening partially determines the rate at which the unit could pump.

2. **Impeller**

   The impeller is that part of the pump which applies energy force to the material pumped to give it velocity and kinetic energy. The opening in the centre of the impeller is called the impeller eye. This is where the flow enters the impeller. The size of the eye partially determines the capacity of the pump.

3. **Impeller Vanes**

   Vanes are blades of the impeller which direct the flow from the eye outward imparting the kinetic energy to the material pumped.

Figures 1-2 and 1-3 illustrate single stage pumps, which are equipped with either one suction inlet (Figures 1-2) or two suction inlets (Figure 1-3).

Figure 1-4 shows a multi-stage pump which may be constructed with either a single or double suction.

In these pumps the fluid is discharged from one stage to the next through internal passages in the pump casing. Each stage builds up the velocity of the fluid until the desired head is developed. A typical multi-stage, single suction pump is illustrated in Figure 1-4.

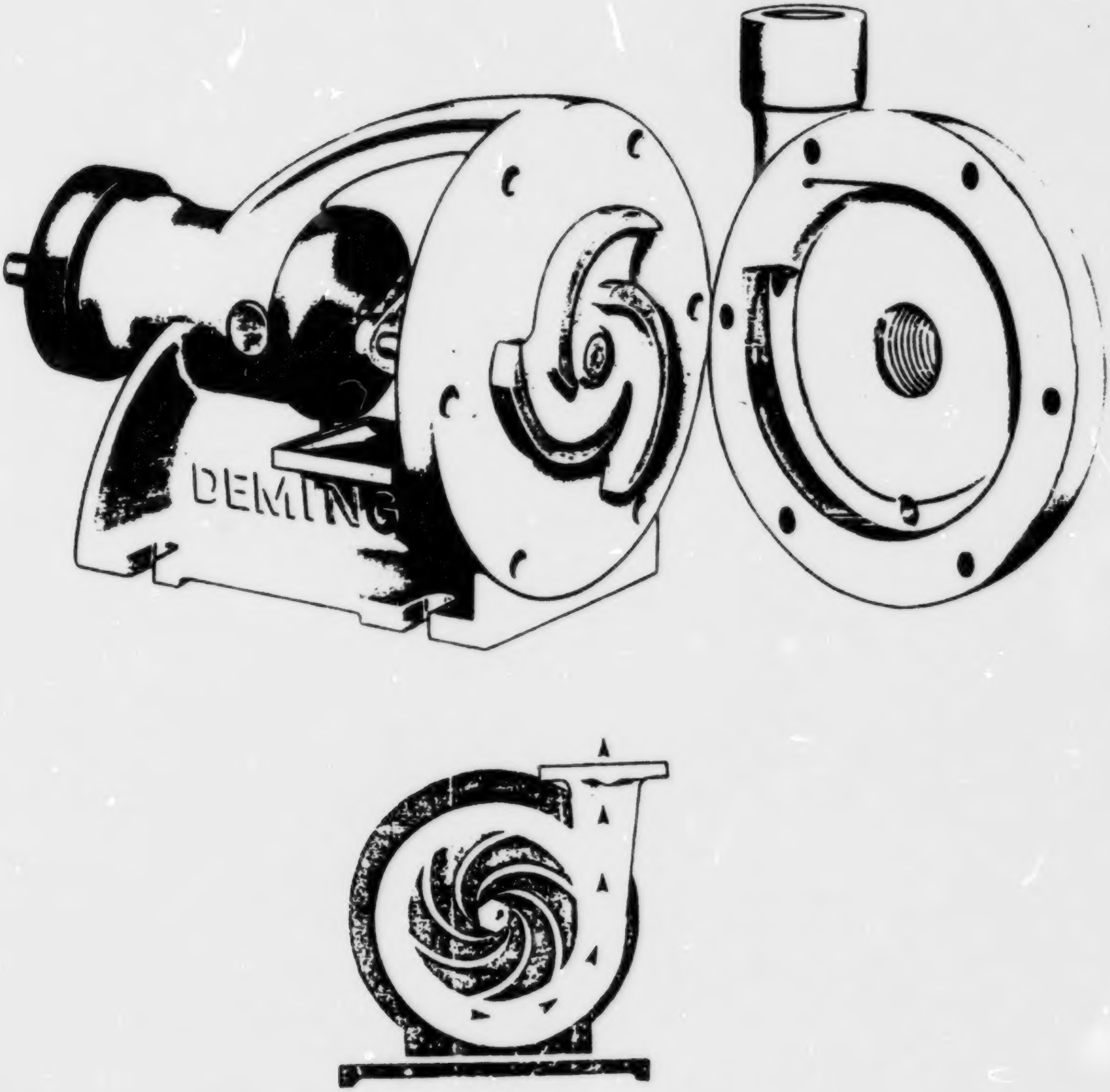


Figure 1-1 End-suction Centrifugal Pump and Diagram Showing the Direction of Fluid Flow.

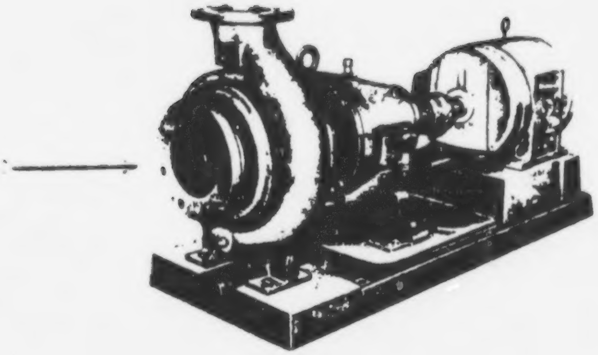


Figure 1-2 Single Stage end-suction Centrifugal Pump.

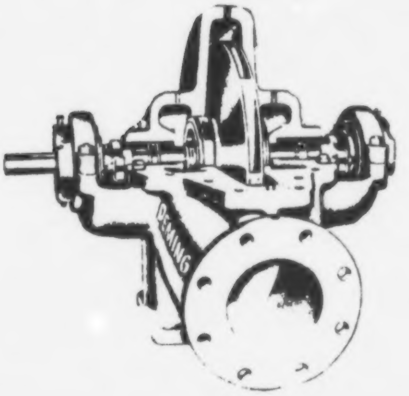


Figure 1-3 Double Suction Pump

4. **Impeller Shrouds**

This component of the pump encloses the blades or the vanes of the impeller, and contains the flow material in the impeller.

5. **Pump Casing**

The casing is the enclosure surrounding the pump impeller, shaft and the packing gland. The casing directs the flow of liquid into and out of the pump. It consists of:

   a) volute
   b) head cover
   c) suction cover

6. **Discharge End**

This is the side or the end where the pump discharges the fluid.

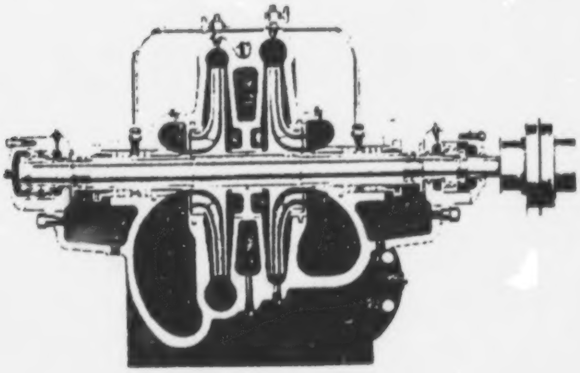


Figure 1-4 Two-stage Single Suction Pump

7. **Pump Shaft**

The pump shaft may consist of a part of a motor, or may be separate to the motor shaft. It may be supported by the motor bearings or the shaft may be supported by independent bearings.

8. **Wearing Rings**

The principal function of the wearing rings is to provide a replaceable bearing surface between the rotating impeller and the stationary casing. This reduces the build-up of heat between rubbing surfaces.

9. **Packing Gland**

The packing gland can be referred to as a seal in a pump. It can be one which uses loose or braided packing or one which has a mechanical seal. Its principal function is to seal the fluid in the pump.

Impellers

The pump impeller is the most important component of the pump. Its size, shape and operating speed determine the pump capacity.

There are two major differences in the shape of various impellers; they are the straight vane impeller and the curved or radial vane impeller. Common types of impellers are shown in Figure 1-5. The straight vane impeller may have a single or a double shroud. The radial vane impeller is also manufactured in both open or closed design. The open-type impellers have the shroud only on the back side. The shroud may cover only part of the vanes or cover the complete diameter of the impeller.

The double shrouded impellers are used in both double and single suction pumps. The double suction-type is very similar to the single suction-type impeller. They differ in that the centre of the double suction impeller has only partial shrouds. The vanes on each side of this impeller are in matching position and join to form a single vane near the discharge edge of the impeller.

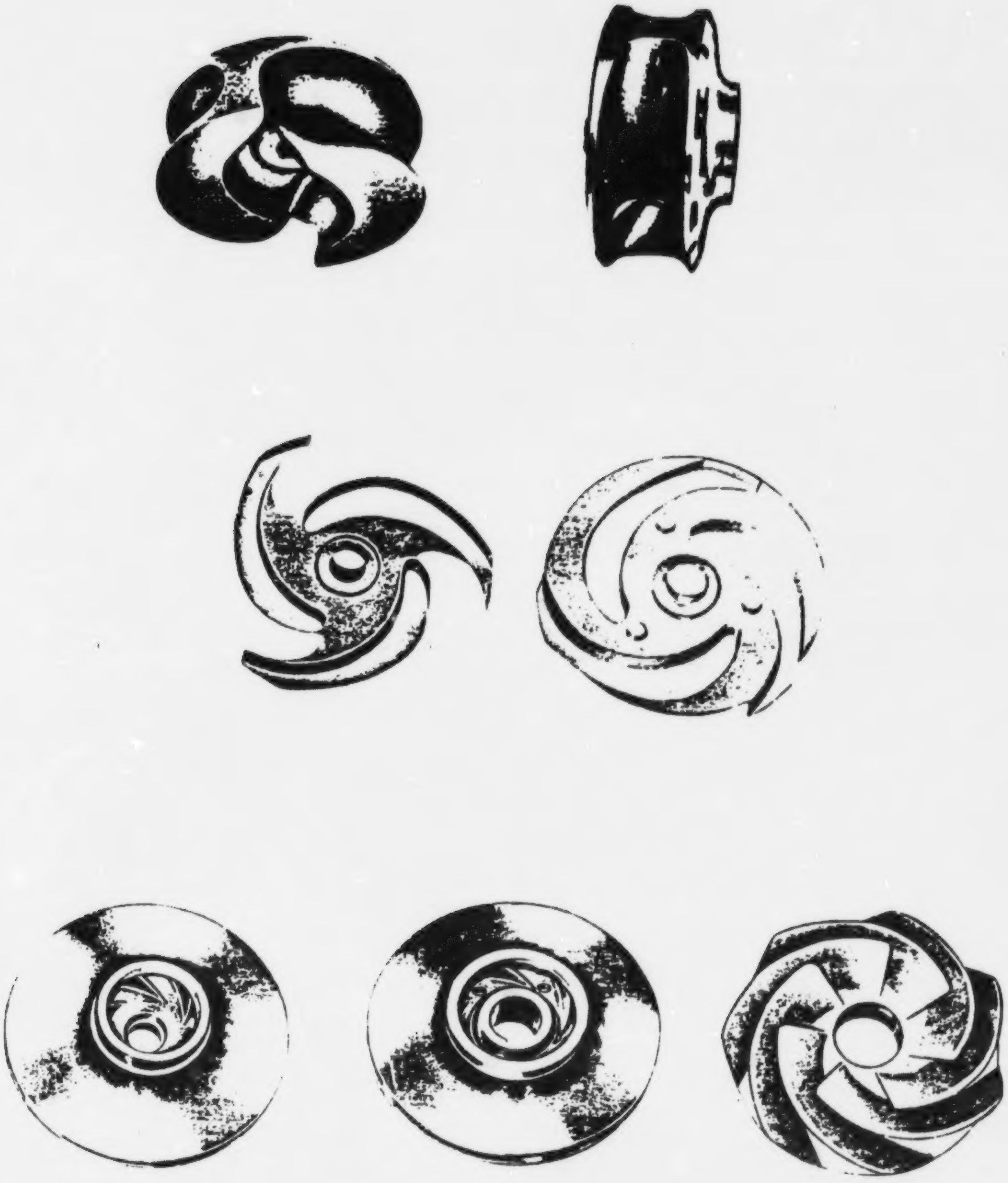


Figure 1-5 Common Types of Impellers

Pump Casings

Centrifugal pumps used in water and wastewater operations vary considerably in design and construction, depending on the application. Some of the distinct physical differences between various types of pumps are illustrated in Figure 1-6.

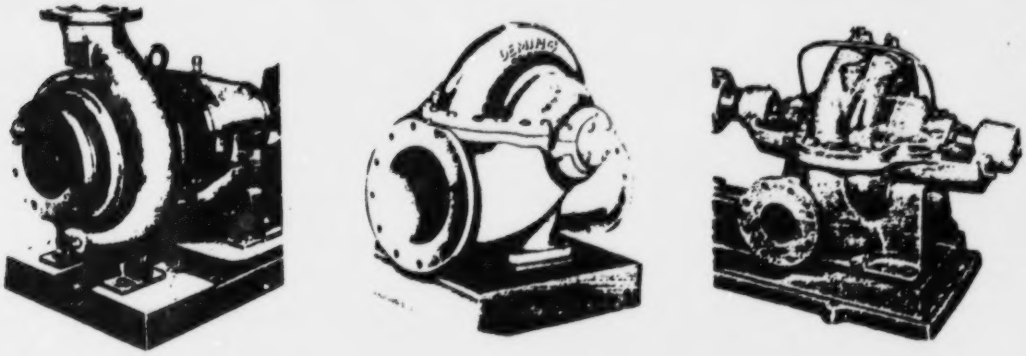


Figure 1-6 Various Types of Centrifugal Pump Casings

Depending on the specific application of the pump, the pump casings are made of different materials such as cast iron, steel, bronze, as well as a number of non-corrosive and abrasion-resistant metal alloys.

Although the general principles of the centrifugal pumps are usually the same, the internal pump design may vary considerably. As an example, in the volute type casing, the internal water action tends to develop unbalanced radial forces, which may be damaging to the pump. To reduce these unbalanced forces, some pumps are constructed with a double volute. This feature provides another guiding vane for the water to pass through, thus splitting and balancing the internal radial forces of the liquid on the pump impeller, shaft and bearings, plus reducing some of the stress load on the casing walls.

The double volute guide vanes can be built into both solid and split-type casings. Also double volute guides are built into multi-stage pumps to balance the discharge head to the suction side of the next stage. A typical double volute casing is illustrated in Figure 1-7.

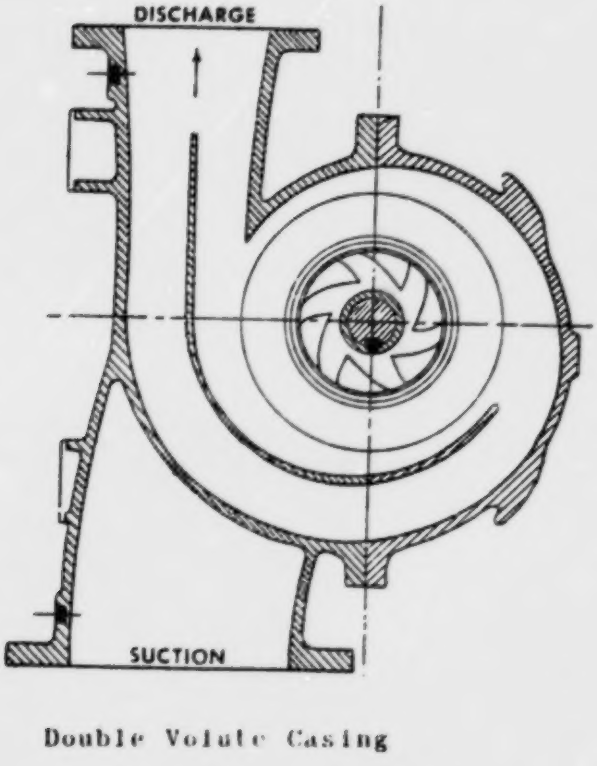


Figure 1-7 Double Volute Casing

In pumps which work against very high heads (pressure), the casings are further strengthened with ribs on the outside of the casing, placed usually at right angles to the direction of the flow.

Pump casings can be of either solid or split design, as illustrated in Figures 1-8 and 1-9. Pumps equipped with solid casings are usually end suction type and are always equipped with a removable cover or face place on the suction side for easy access to the impeller.

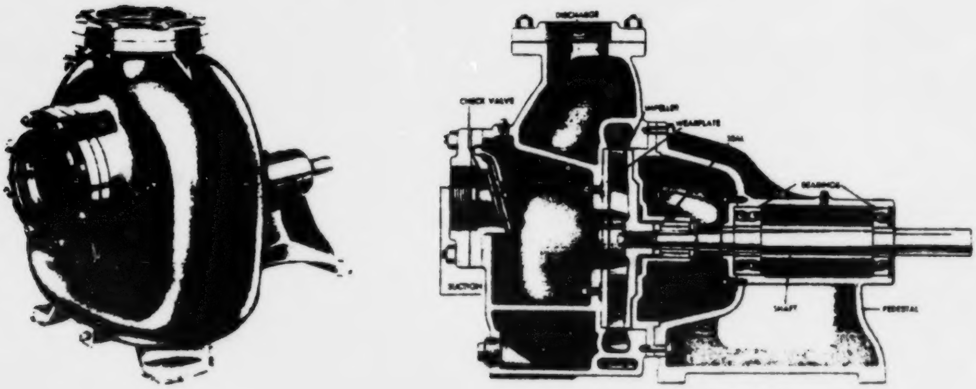


Figure 1-8 Solid Casing Design

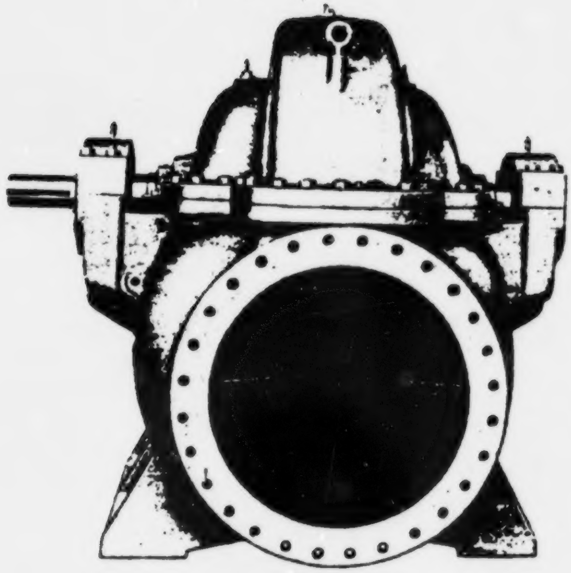


Figure 1-9 Split Casing Design

Pump casings consisting of split construction are normally called the horizontal or axial split. This type of design with a split on the centre line of the shaft allows for the removal of the upper half of the casing and provides access to the pump shaft, impellers and bearings.

Pump casings are also manufactured with a radial split. With the radial split, the casing can be assembled in segments to form one multistage pump shown in Figure 1-3.

Pumps which are cast complete with a supporting base usually have the direction of flow, horizontally or vertically, predetermined by its design. A pump which requires independent supporting framework may have its discharge point positioned in a number of different ways to fit in with the installation requirements.

Double suction pumps are normally constructed with both the intake and the discharge below the centre line, Fig.1-10. This pump can be equipped with the intake and discharge on opposite sides, or it may have an intake at the bottom and have a side discharge port.

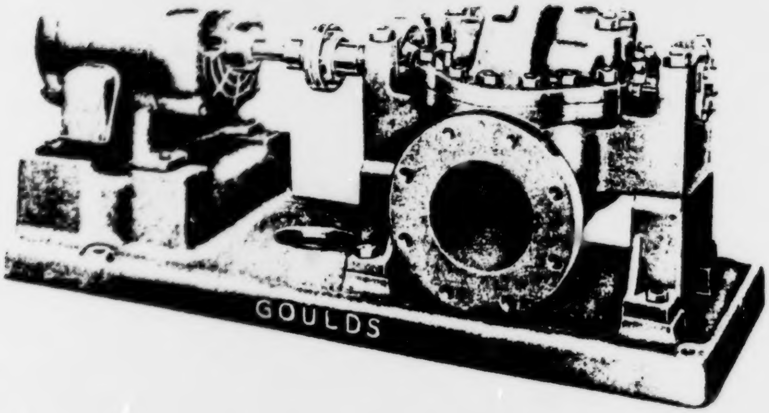


Figure 1-10 Double Suction Pump Casing

Wearing Rings

The wearing rings in the centrifugal pumps serve the following purposes:

1. It is a replaceable bearing surface between the rotating impeller and the stationary casing.
2. It reduces the heat between rubbing surfaces.
3. It provides a leakage point for fluid between the discharge and the suction side of the pump, allowing small amount of water to recirculate inside the pump. This recirculation counteracts some of the radial and thrust forces that build up within the pump. In most cases the wearing rings are designed to be adjustable so that a proper clearance between the impeller and the casing could be maintained.

The wearing rings are normally made of bronze or brass. They may be installed either in the casing or on the impeller, or on both depending on the design of the pump.

Pump Shafts Bearings

The shaft bearings, depending on the design of the pump, may be of sleeve, roller bearing or ball bearing type. Also, some types of pumps are equipped with special thrust bearings to eliminate the development of high thrust against the end of the pump. Multistage pumps with high pressure capacities are usually equipped with thrust bearings.

In low horsepower pumps, the impeller and the motor are on the same shaft. Therefore, in this case all radial and axial thrust must be absorbed by the motor bearings.

Large pumps usually have separate shafts and are coupled to the motor. The shaft is independently supported by its own set of bearings according to its design (Fig.1-11).

Typical coupling are shown in Figure 1-11.

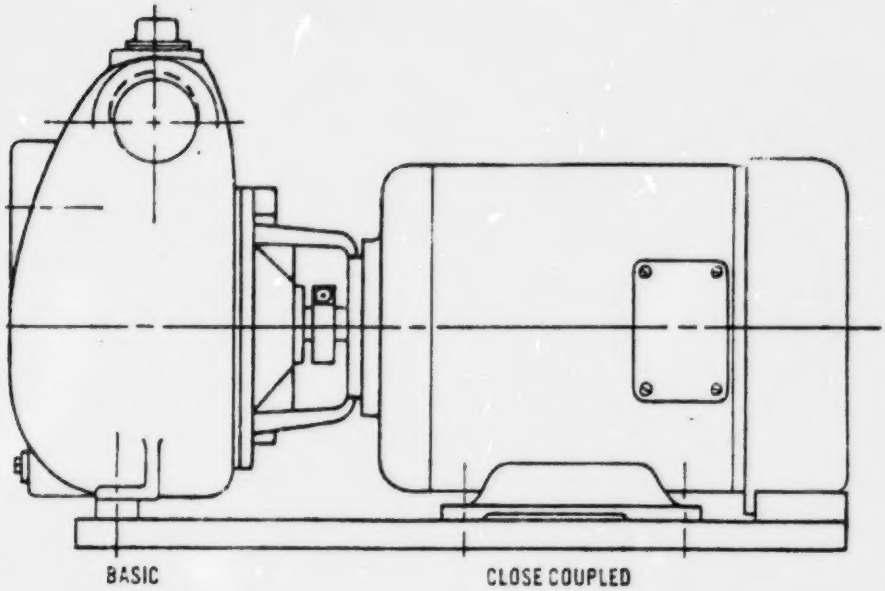
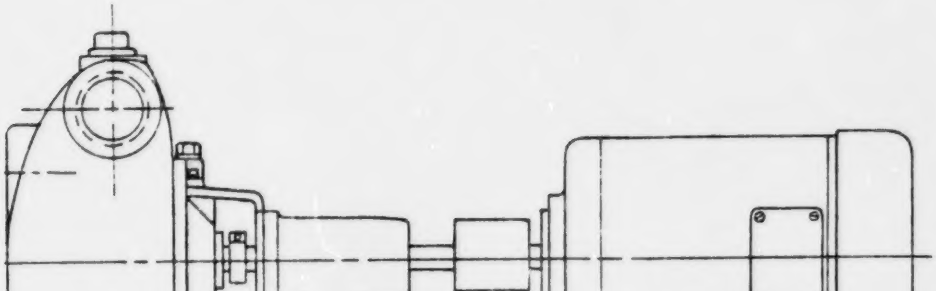


Figure 1-11 Typical Pump to Motor Couplings



PROPELLER AND TURBINE PUMPS

General

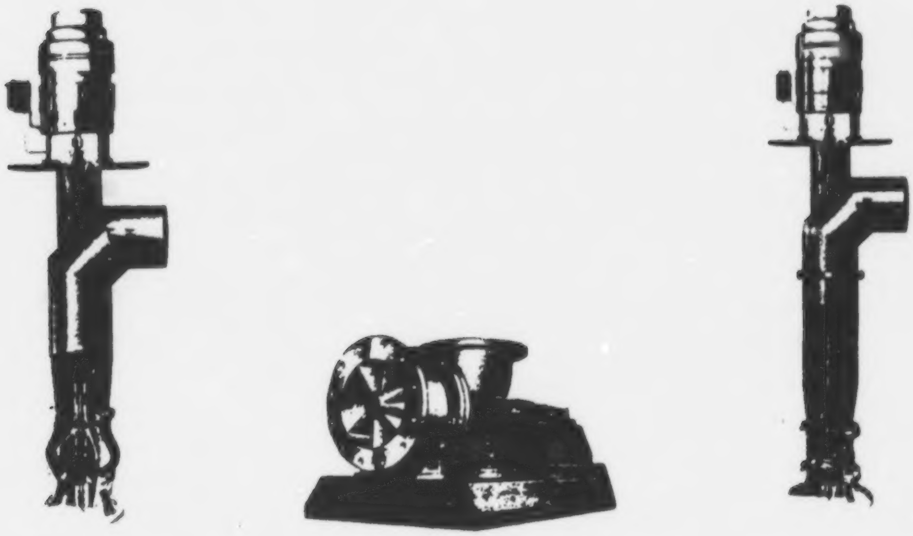
Although the volute type pump is the most common pump used in municipal water and wastewater operations, there are instances where this pump is not capable of delivering the required capacity or providing the required lift. For these applications either the propeller or turbine pump is selected.

The propeller pumps are classed as **Axial-Flow** or **Mixed-Flow**.

Axial-Flow Pumps

Axial-flow pumps are designed for installations where high volume and low head capabilities are required. This type of pump is often used in municipal water supply systems, or for irrigation purposes or dewatering of flooded areas. The pump may be built with either below or above grade discharge points. Some typical pumps are shown in Figure 1-12.

The propeller casing of this pump frequently resembles a short length of pipe as compared to special design or construction of the centrifugal pump. Usually the pump casing is equipped with a set of **Diffuser Vanes**. The purpose of these vanes is to reduce the turbulence caused by the action of the impeller and also to direct the flow of water to the discharge pipe.



Mixed Flow Pump

Horizontal Propeller Pump

Vertical Propeller Pump

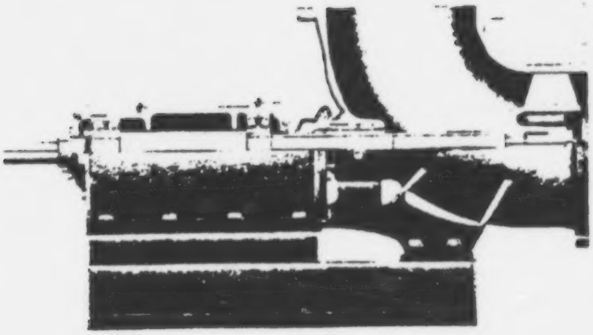


Figure 1-12 Typical Axial Flow Pumps

Mixed-Flow Pumps

Although this pump is very similar in design to the axial-flow, the impeller heads are quite different, as illustrated in Figure 1-13.

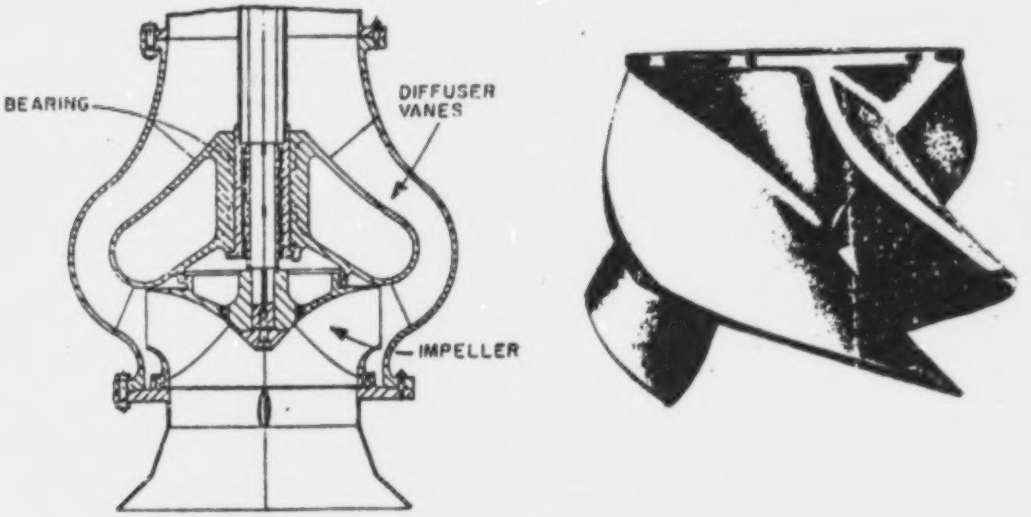


Figure 1-13 Impeller for the Mixed-Flow Pump

Unlike the axial-flow impellers which move the water in straight lines parallel to the shaft, the mixed-flow impellers give the fluid a swirling or a slight centrifugal motion as it leaves the impeller blades. This design provides the pump with a capability of a slight suction lift. However, these pumps are restricted to operating speeds of above 4200 rpm. Operating at speeds below 4200 rpm decreases the efficiency of the pump considerably.

Although certain mixed-flow pumps resemble a vertically mounted centrifugal pump, it is designed to handle much greater volumes of water.

For installation in wells, the casing of the axial-flow or mixed-flow pump is built slightly smaller than the discharge piping. This is to allow for the removal of the pump for servicing without the necessity of having to dismantle the well piping and the supports. The pump casing is connected to the discharge pipe by a split adapter ring. When this ring is removed, the pump can then be lifted without any difficulty.

The vanes on the mixed-flow impeller are enclosed in a shroud discharging the fluid outwardly much the same as in the centrifugal pump.

With exception of the trust bearing, the bearings supporting the shafts in axial-flow and the mixed-flow may be of the sleeve, roller or ball type. They can be lubricated by grease, oil or water depending on the application or the pump's design. Depending on the manufacturer's specification, the pump seals can be either mechanical or packing type.

Vertical Turbine Pump

Vertical turbine pumps are often used in municipalities to pump water from deep wells. When necessary, this type of pump can be designed to work under head pressures of 1,000 feet. The size and capacity of the pump is usually restricted by the diameter of the well and the amount of water it is capable of supplying. This pump is usually built up of a number of segments or stages. The number of segments required will depend upon the well-head requirements, Fig.1-14.

The impeller casing is usually called the pump bowl. The height of the bowl may vary with design of the pump. In some pumps the impellers will be fairly flat, while in others they will be shaped similar to the mixed-flow impeller.

Bearings are normally installed in each casing bowl. This means that in pumps with multiple series each bowl has separate bearings supporting the shaft. Bearings are also located at intermediate points along the drive shaft rising to the motor. In water supply installations, the bearings are lubricated by water being pumped.

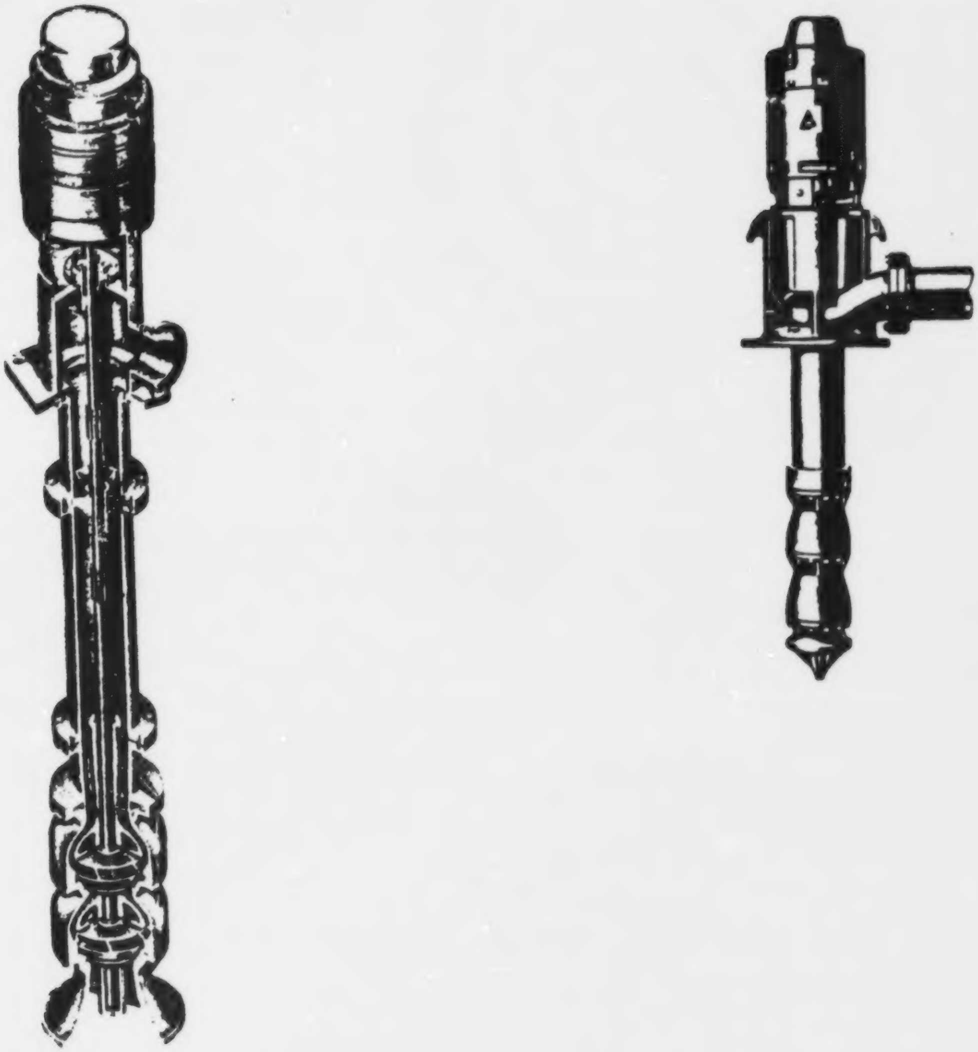


Figure 1-14 Typical Vertical Turbine Pump

POSITIVE DISPLACEMENT PUMPS

ROTARY PUMPS

General

The term Rotary Pump includes a wide variety of pumps designed for handling:

(a) liquids

(b) vapour-liquids

(c) gases

They are available in sizes from small cooling-water circulating to fairly large units handling viscous products.

Rotary pumps are positive displacement pumps, with relatively steady (non-pulsating) flow. The capacity is approximately proportional to the pump speed. Capacity control of this type of pump may be obtained by varying the speed of the pump or by recirculating a portion of the discharge.

Liquids with a wide range of viscosity can be handled by rotary pumps. They will operate satisfactorily on liquids of high viscosity, where centrifugal pumps will not function.

Rotary pumps are self-priming and can handle a certain amount of vapour or gas together with the liquid pumped. Certain types of these pumps, when properly lubricated, can be used to pump gas or vapour continuously as vacuum pumps.

Types

Most rotary pumps are of three general types:

1. gear pumps
2. vane pumps
3. eccentric-rotor (cane and piston)
4. screw pumps.

These pumps are shown in Figure 1-15. The gear and screw type pumps may be classified as:

(a) internal bearing
(b) external bearing.

Certain gear and screw pumps are fitted with external timing gears which prevent contact between the pumping rotors.

The internal bearings are lubricated by the product pumped, while the external bearings are lubricated separately in a housing separate from the pump case by stuffing boxes. Figures 1-16(a) and 1-16(b).

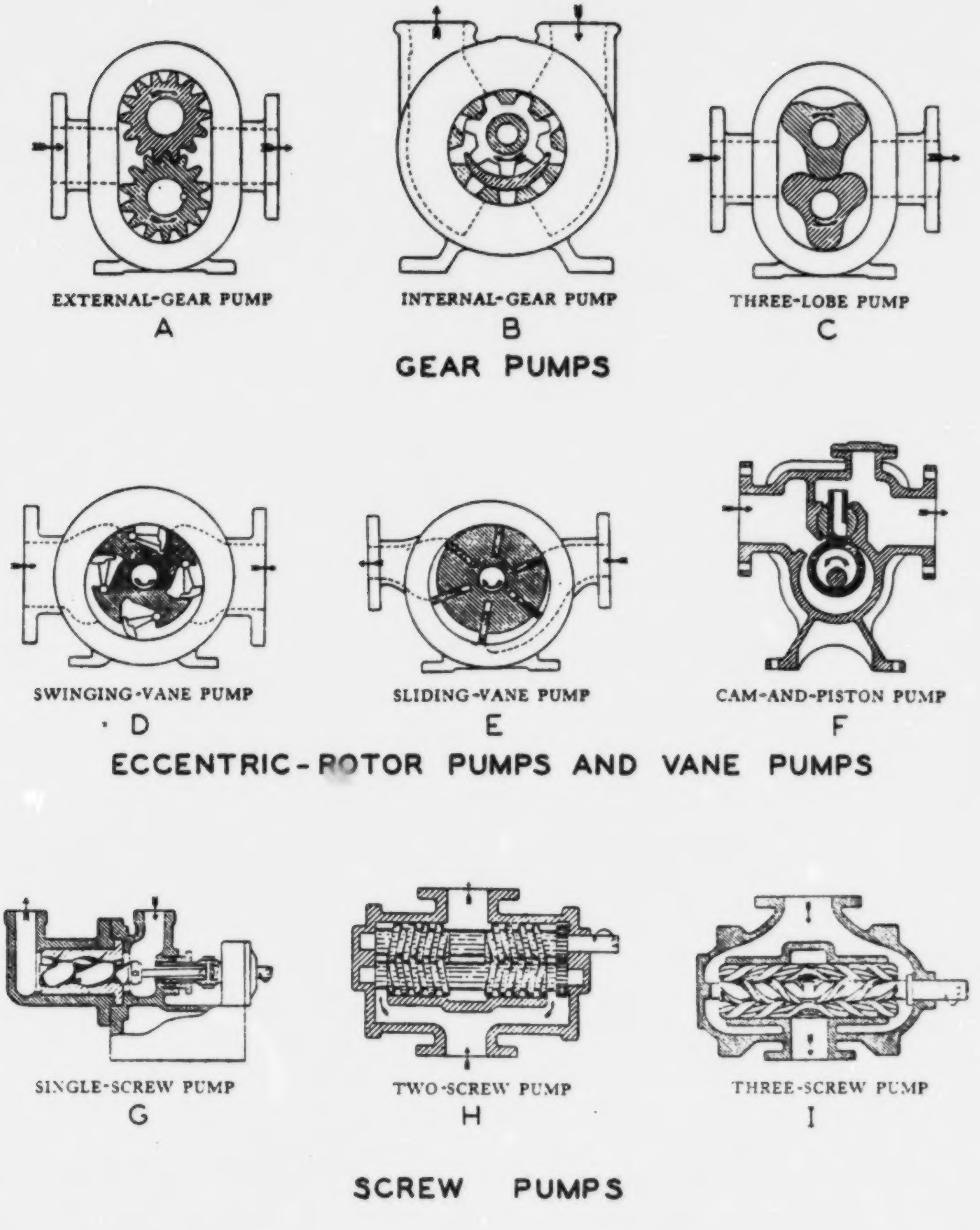
If cavitation is to be avoided, rotary pumps handling liquids must have sufficient pressure at the suction end to ensure that they are completely filled with liquid. The suction pressure required depends on pump speed, design and on the viscosity of the material pumped. Rotary pumps will work satisfactorily with low NPSH provided the pump is not operated at an excessive speed. If NPSH available is lower than required, the speed of the pump must be reduced.

When viscous materials are handled, care should be taken in sizing the suction piping to ensure that the required flow rate could be delivered to the pump. If NPSH is limiting and the viscosity is increased, the size of the suction passages must be increased or the capacity of the pump must be reduced by reducing the speed of the pump so that friction drop in the suction passages will not prevent the pump from filling completely.

Relief Valves

Since rotary pumps are positive displacement pumps, pressure relief valves are required to protect them from over-pressure. In some designs, a recirculating relief valve is installed as an integral part of the pump. This valve should be inspected periodically to ensure that it is not crusted up and stuck in closed position.

The features of rotary pumps are summarized on page 01-32.



EXTERNAL-GEAR PUMP
A

INTERNAL-GEAR PUMP
B

THREE-LOBE PUMP
C

GEAR PUMPS

SWINGING-VANE PUMP
D

SLIDING-VANE PUMP
E

CAM-AND-PISTON PUMP
F

ECCENTRIC-ROTOR PUMPS AND VANE PUMPS

SINGLE-SCREW PUMP
G

TWO-SCREW PUMP
H

THREE-SCREW PUMP
I

SCREW PUMPS

Figure 1 15 Common Type Rotary Pumps

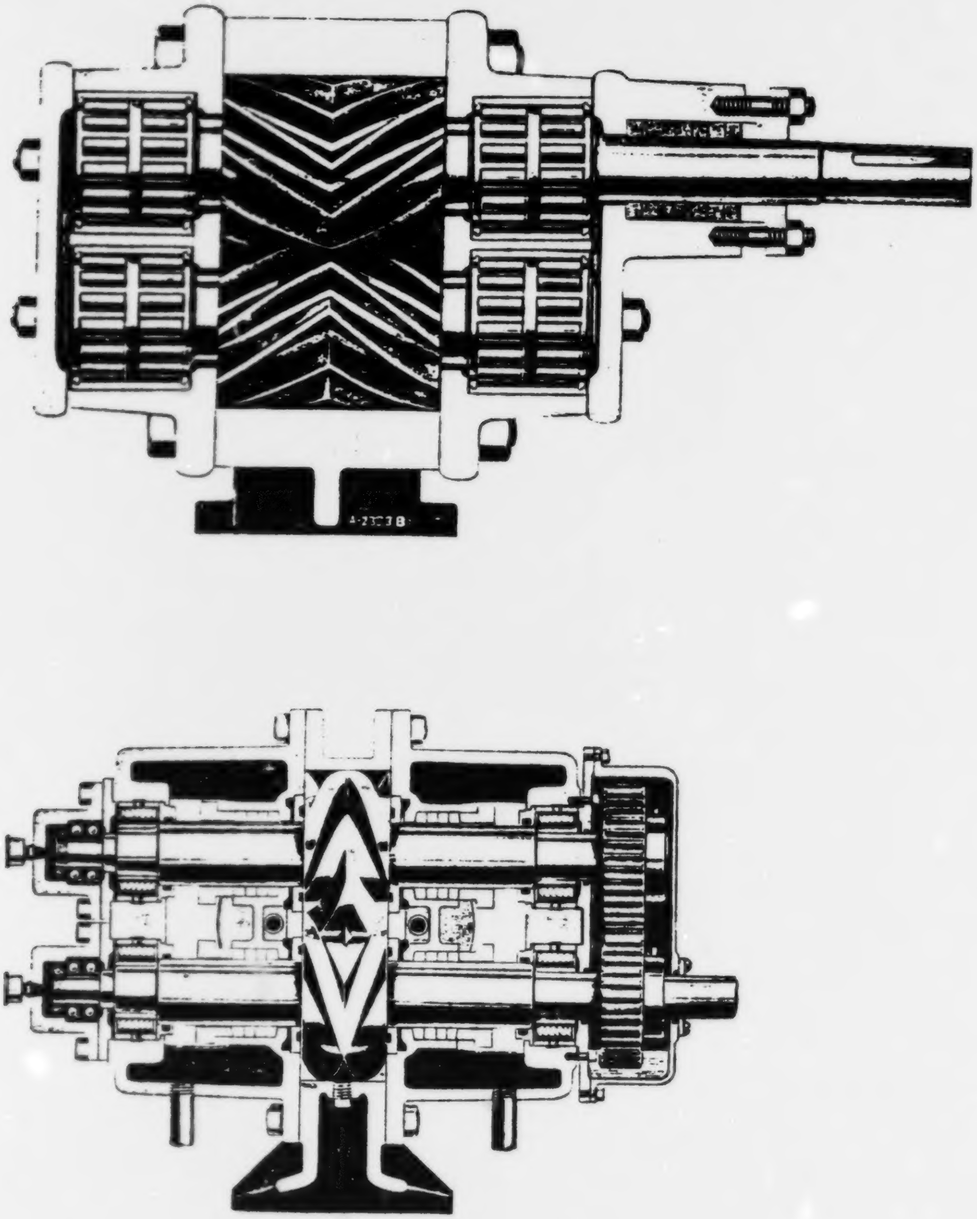


Figure 1-16 (a) Gear type Rotary Pump with internal bearings

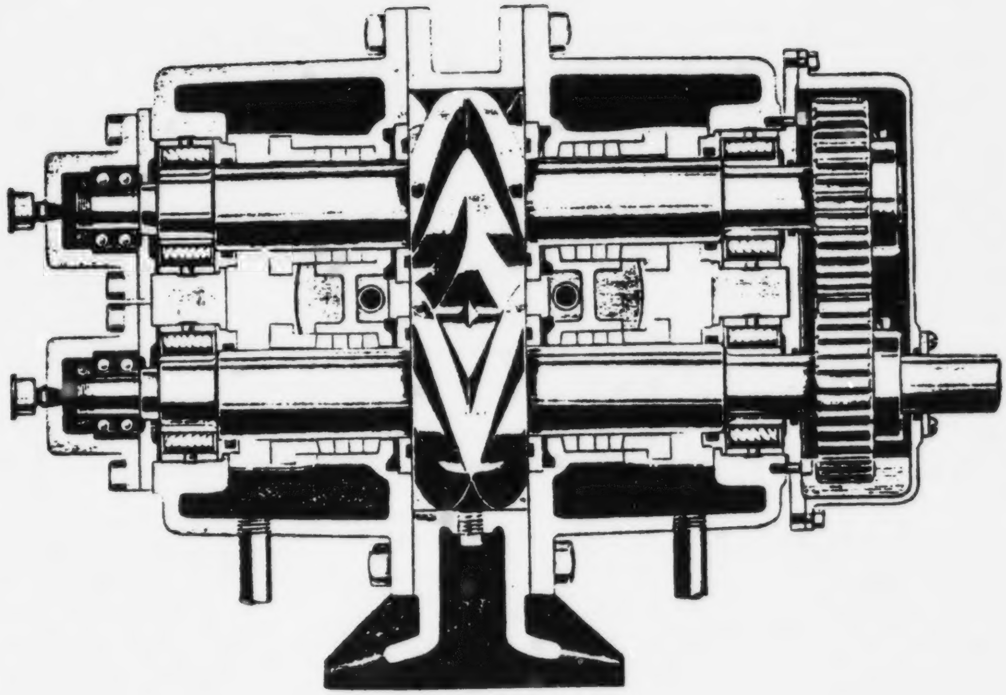


Figure 1-16 (b) Gear-type Rotary Pump with external bearings.
Courtesy of Worthington Corporation

FEATURES OF ROTARY PUMPS

(Chart)

FEATURES OF ROTARY PUMPS

	Gear Pumps	Eccentric-Rotar Pumps and Vane Pumps	Screw Pumps
Frequency of use	Generally least expensive and most frequently used.	Often inexpensive but excessive wear except on clean lubricating stocks may limit application. Frequently used for handling air or gas.	Often only type available in higher pressures for larger capacities. Usually more expensive than other types in smaller capacities and lower pressures. Almost a standard for delivering fuel oil to burners.
Internal Design	Various forms of gears are used: external spur, internal spur, herring-bone, and spiral.	Eccentric rotor or case with vanes or other sealing members. Always has metal-to-metal rubbing, surfaces lubricated by product pumped. Some types have removable liners and vanes which can be replaced when worn.	Meshing screws. Thr rotor of the single-screw pump does not run concentrically but circles around the axis of the stator.
Speeds	Herringbone and spiral gears can be run at standard motor speeds of 1150 and 1750 rpm in smaller pump sizes. Larger sizes and other gear types ordinarily require lower operating speeds.	Usually slow speed requiring geared drivers. Small capacity units sometimes operate to 1750 rpm.	In single-screw and two-screw pumps, speeds up to 1750 rpm are obtainable in smaller sizes and for some services. Lower operating speeds are usually required. In three-screw pumps motor speeds of 1750 or 3550 rpm are usually acceptable, and, for occasional services, speeds to 5000 rpm are satisfactory.
Bearing Location and Lubrication	Either internal (lubricated by product pumped) or external (separate housing for separate lubrication). See Figures 3-2 and 3-3.	All bearings and rubbing surfaces internal (lubricated by product pumped).	Either internal or external bearings for two-screw pumps. External bearings on drive shaft of single-screw pump but there is rubbing contact between stator and rotor. Internal only for three-screw pumps.
Timing Gears (to prevent contact between rotors)	Used in external-bearing designs or when special gear teeth shapes do not allow one pumping gear to drive the other.	None	Required for two-screw pumps. Not available on single- or three-screw pumps.
Working Pressures	Normally up to about 400 psi with internal bearings. Special design to about 2000 psi for filtered lubricating liquids. For external bearings pressures to 300 psi are obtainable in small sizes, with 50 to 150 psi for larger sizes.	Varies with design and size. Pressures to 1000 psi are obtainable but most common designs are for 75 to 200 psi.	Single-screw pumps to 200 psi. Two-screw pumps to 400 psi. Three-screw pumps to 1500 psi.

RECIPROCATING PUMPS

Reciprocating pumps are positive displacement pumps. Within the limitations of pump construction, the maximum pressure developed is limited only by the torque of the driver for crank-driven pumps. The capacity can be controlled by varying the speed of the driver and adaptable speed reduction gears.

The flow of liquid from this type of pump is pulsating, varying both in flow rate and pressure. One of its principal applications in water and wastewater treatment is for handing materials which are highly viscous or contain heavier solids. A cross-section of the pump is shown in Figure 1-17.

Types

Following are a few of the most common means by which reciprocating pumps are classified:

1. Type of Drive
    - (a) Direct acting, steam driven
    - (b) Crank-driven (often called "power" pumps) (Figure 1-17)
2. Cylinder Orientation
    - (a) Horizontal
    - (b) Vertical

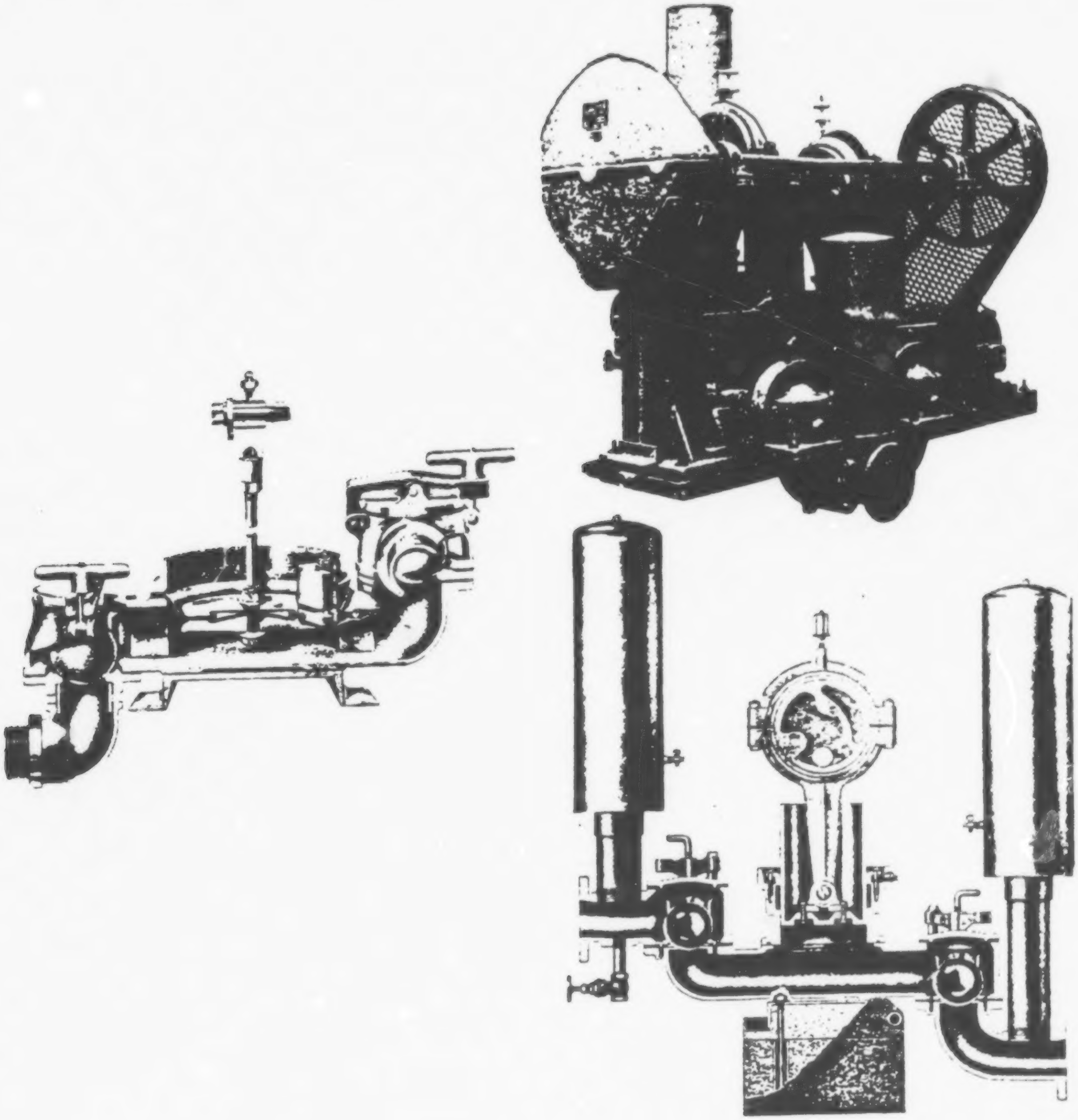


Figure 1-17 Typical Reciprocating Pumps

3. Liquid-end Arrangement
    - (a) Plunger-type (outside packing)
    - (b) Piston-type (inside piston rings or packing on the piston)
4. Valves and Cylinders (classified by arrangement and design)
    - (a) Valve-plate
    - (b) Pot valve
    - (c) Turret-top
    - (d) Close-clearance
    - (e) Forged-steel, integral valve.
5. Number of Pistons or Plungers
    - (a) Simplex
    - (b) Duplex
    - (c) Triplex
    - (d) Quintuplex, etc.
6. Type of Action
    - (a) Single-acting (delivers on either forward or backward stroke, not both).
    - (b) Double-acting (delivers on both forward and backward strokes).

Sizes and Size Designation

The sizes of reciprocating pumps vary from small proportioning pumps with very short adjustable strokes to large high- pressure pipe line pumps with strokes of 36 inches or more.

The size of a simple reciprocating steam pump is expressed in inches by three figures as follows: diameter of steam cylinder x diameter of liquid cylinder x length of stroke (e.g. 6 x 5 x 6). Note that this size designation does not tell whether the pump is simplex or duplex, valve plate or side pot, close-clearance or not, etc.

The size of crank-driven reciprocating pumps is usually expressed in inches with two numbers as follows: diameter of the liquid piston x length of stroke (e.g. 3 x 10).

Cylinder Liners

Most reciprocating pumps are equipped with cylinder liners. There are two general types - replaceable and removable. Replaceable liners are press-fit and must be removed with a cutting tool and replaced in a hydraulic press. Removable liners are usually held in position by bolts or screws and can ordinarily be removed with hand tools. Removable liners are preferred.

Valve Arrangements

The most common valve arrangements for the liquid end of reciprocating pumps and their approximate pressure limitations are:

1. Valve-plate construction provides a single cover for all the suction and discharge valves. This construction is not commonly available for discharge pressure above about 250 psi in cast iron.
2. Pot-valve construction provides each suction and discharge valve with a separate chamber usually cast on the side or top of the cylinder. Each "pot" has a separate bolted cover and can be designed for high pressure.
3. Integral-valve construction provides a separate chamber for each suction and discharge valve machined in the cylinder block. This is the usual construction for forged steel liquid ends and is suitable for the highest pressures for which reciprocating pumps are designed.

Valve Types

The type of valve for a given service is usually determined by the pump manufacturer. Some of the most common types of valves are:

1. A disc-type valve is simply a flat plate which closes over the valve port, usually under spring pressure. Disc-type valves of special design are also used on high-pressure, high-speed reciprocating pumps for pipe line and similar services. These valves have operated satisfactorily at pressures up to 2,000 psi.
2. Wing-guided-type valves have an extension below the seat of the valve to provide a guide which keeps the valve in line as it rises and assures its seating properly. Wing-guided valves are ordinarily offered for pressures above 350 psi and are applied up to the highest pressures for which reciprocating pumps are designed.
3. Ball-type valves are round balls which usually seat themselves in a different position each time they open and close, thus preventing continued wear on any one spot. Ordinarily, ball valves or semi-ball valves of the "Rollo" type offer less resistance to flow and so are recommended for pumps handling highly viscous materials. They are, however, comparatively heavy and therefore not well adapted to high-speed service.

Length of Stroke

Other factors being equal, a long stroke with a piston or plunger of comparatively small diameter is usually more satisfactory than a shorter stroke with a larger piston or plunger diameter. The reduced shaft or plunger size gives longer packing life and the efficiency is usually improved as the length of the stroke is increased. However, the very high speed pumps that have recently come into fairly wide use have a short stroke. Their frames, bearings, and shafts are very heavy and rigid.

Controlled Volume Pumps

In water and wastewater operation, a requirement frequently arises for a relatively small capacity, high-discharge pressure pump to handle chemicals at controllable rates. Generally controlled-volume reciprocating pumps, often called proportioning pumps, are most satisfactory. If the chemicals handled are corrosive to iron or steel, special materials must be used.

Controlled-volume pumps are designed for ready adjustment of the flow rate. They could deliver the set flow against varying discharge pressures. These pumps are usually reciprocating plunger pumps of the single action crank driven type.

Diaphragm-type, controlled-volume pumps are also available and eliminate the stuffing box that may be a source of trouble in the plunger-type pumps (Figure 1-18).

These pumps are manufactured to handle capacities as low as a few cubic centimetres an hour up to about 25 or 30 gpm. In the lower capacities differential pressures up to 20,000 psi are available. Typical services include:

(a) Injection of acid or other chemicals for water treating.

(b) Controlled injection of chemicals into process streams.

The flow rate in these pumps is controlled by varying either the stroke length or the pump speed. Where necessary, these pumps can be fitted with controls permitting flow adjustments while the pump is running. This adjustment can be automatically controlled when process conditions require.

Piston Pumps vs Plunger Pumps

Piston pumps are almost invariably cheaper than plunger or outside-packed pumps. The design of plunger pumps, however, allows injection of an external lubricant into the stuffing box if needed to prolong packing life. Further, leakage past the packing can be observed and the packing adjusted as required.

None of this is possible with the internally packed piston-type pumps. Piston pumps, however, with metal piston rings for packing are available in crank-driven types for pressures up to 1,500 psi.

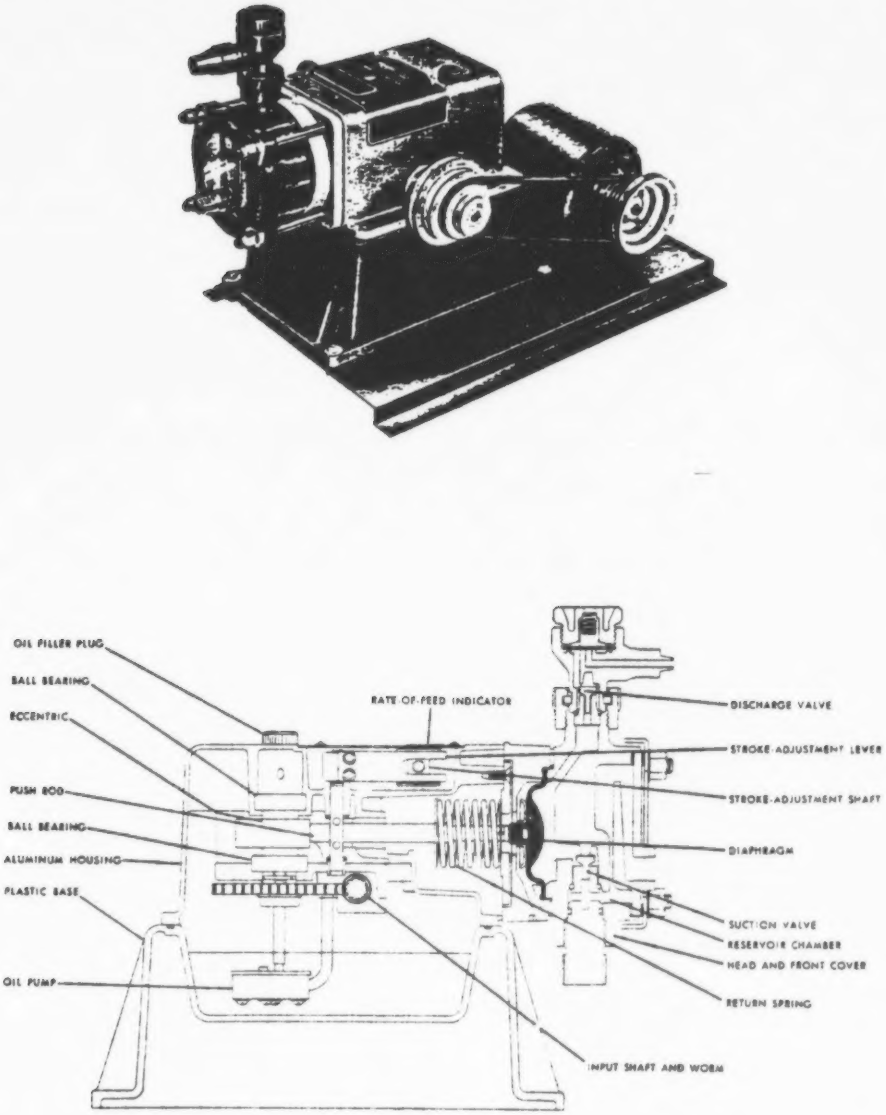


Figure 1-18 Diaphragm type Chemical Feed Pump

Applications of Reciprocating Pumps

In recent years reciprocating pumps have been replaced in many applications by centrifugal pumps largely because of cost. Reciprocating pumps, however, possess advantages of flexibility of head and capacity with nearly uniform efficiency over a wide range of conditions. Thus, for certain services, they are a better selection than centrifugals. In general, they are preferred for small capacity high-pressure services.

While small reciprocating pumps are fairly competitive with centrifugal pumps, in large sizes (particularly over 200 gpm capacity) they are apt to be more expensive than other types.

For many services, the operating costs of motor-driven or turbine driven centrifugal pumps will be less than for reciprocating pumps. Maintenance costs for reciprocating pumps will usually exceed those of a centrifugal pump because of the valves, the many moving parts, and the sliding contacts in the reciprocating pump.

Pulsating flow sometimes limits the application of a reciprocating pump. It will seldom be found, however, that pulsating flow alone is a controlling factor for services where reciprocating pumps are otherwise the best selection. The effects of pulsation could be corrected by the use of pulsation dampers in certain installations. Further, in some cases pulsation effects can be minimized by selection of adequate line sizes since the magnitude of the pressure variation is inversely proportioned to the flow area of the pipe. This is particularly important for low speed pumps discharging into relatively short lines.

Pulsating flow may introduce some problems in the application of automatic control and flow measurement, but usually this can be taken care of without undue difficulty or cost.

Some usually good applications are:

1. Low capacity with relatively high differential pressure or moderate capacity with very high differential pressure.
2. Relatively high viscosity. Reciprocating pumps can economically handle stocks up to about 8,000 SSU (both viscosities at flowing temperature). However efficiency of a centrifugal pump drops rapidly with increasing viscosity.
3. Relatively constant capacity with widely varying discharge pressures. Crank-driven pumps are particularly suited to this application.
4. Highly variable capacity with either constant or varying discharge pressure.

Some usually poor applications for reciprocating pumps are:

1. Whenever pulsating flow is in fact objectionable. An example of such a service is the feed of fuel to boilers. Ordinarily, rotary pumps are preferred for this service because of their smooth discharge pressure.
2. Medium capacity and medium differential pressure with low viscosity. Centrifugal pumps will ordinarily be more economical. Examples of such services are water or petroleum products in the range above about 100 gpm with differential pressures up to about 300 psi.
3. High capacity. For capacities beyond about 200 gpm, reciprocating pumps are rarely the proper selection regardless of the discharge pressure. In this range the size of reciprocating pumps becomes so large that they are more expensive than centrifugal pumps.

SUBJECT:

PREVENTIVE MAINTENANCE

TOPIC: 2

UTILIZATION OF DRAWINGS AND EQUIPMENT MANUALS

OBJECTIVES:

The trainee will be able to:

1. Recall general information available in the manufacturer's equipment manual.
2. List the important factors found under the following categories in an equipment manual:
   (a) Installation Procedures
   (b) Start-up procedures
   (c) Service and repair
3. Recall the use of electrical and piping symbols used on schematics.
4. Read plans by use of the front view, side view, top view and sectional.
5. Interpret the legend on drawings including the unit of measurement and the graphic symbols.
6. Demonstrate the use of:
   (a) construction drawings
   (b) mechanical drawings

UTILIZATION OF DRAWINGS AND EQUIPMENT MANUALS

INTRODUCTION

An operator may be confronted with the need to examine drawings of his plant. If he has a broken pipe and has to locate it, memory may fail; where an accurate set of drawings, IF AVAILABLE, will provide him with the shortest and most economic distance between him and the problem. He can learn location, depth, type, size and perhaps a clue to his problem because of the configuration of the original installation. When a plant is being expanded, the operators' opinion on directional flow, type of valve required or other operating criteria may be asked for. At the drawing stage is the time to voice opinions. After that one may be stuck with a problem, one that an operations person would appreciate at the drawing stage and could have eliminated before it became an expensive and troublesome fact.

Further, where are the instruction manuals for the plant equipment? Is memory serving as the guide to schedules for oiling and greasing equipments? Can parts which need replacing, and the supplier of the original material be easily identified? How often is a new piece of equipment installed and put into operation according to the manufacturer's start-up procedures? Is the instruction manual used or is it filed and forgotten? Proper use of the equipment manual will ensure satisfactory operation as well as preserve the integrity of the manufacturer's warranty.

This topic will discuss the information available from drawings, manuals and other reference material and the importance of making use of these sources.

EQUIPMENT MANUALS

Accessibility

Where are the equipment manuals? If they are not readily available, one of the most useful tools in plant operation is lost. First of all, locate a manual if one is printed for each piece of equipment in the plant. This may require searching through piles of debris or sending to a manufacturer requesting its replacement. In any case, obtain them and file them by some orderly system so that when one is required it is available.

An equipment manual is a textbook on the equipment one depends on to produce a satisfactory product. If the fundamental requirements for the operation of a piece of equipment are known, a basis for its proper care is established. The use of this information coupled with operating experience should provide high quality maintenance control.

An equipment manual can offer information on:

1. Identification of the unit and supplier
2. Installation
3. Start-up
4. Operation and maintenance
5. Troubleshooting
6. Service and repair

Installation

Minor pieces of equipment can malfunction as easily as major ones. When a replacement for a pump is required the reasons for the failure of the original equipment should be examined. Was it caused by the site location or the way it was secured? Was it caused because the recommended procedure from the manual is wrong or was not followed? Were short cuts taken during installation? Know what is involved from foundation to securing requirements to ensure that the installation is according to design. It will save a lot of grief, time and money if installation is correct from the start.

Start-Up

Even if a new pump is properly installed it is important that the manufacturer's start-up and operating procedures be followed. If, due to local conditions, it is not possible to adhere to the manufacturer's instructions, it would be advisable to obtain the manufacturer's written approval of the amended procedures. Failure to do so may, in the event of trouble, invalidate the warranty. Always use the manufacturer's start-up check list.

Service and Repair

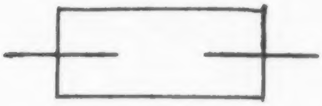
The manual will provide considerable information for service and repair of the equipment. Often, detailed drawings will be present indicating specific parts for ease of identification in ordering from the supplier. Frequently, drawings may be general drawings and represent several models of the same design. Be certain the model and serial number is obtained directly from the unit so that the correct section of the manual is used for obtaining the required information. Details of areas of frequent repair may be covered thus providing a list of replacement parts, to be kept in stock. This can offer the advantage of minimizing the down time of the equipment.

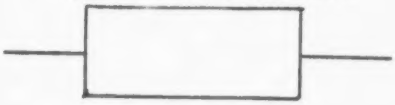
SYMBOLS

A symbol is a standard diagrammatic representation of a piece of equipment. Its purpose is to simplify the illustration of the components in a drawing and still indicate accurately the components purpose.

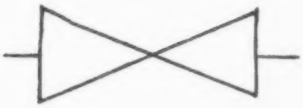
On the following pages are copies of standard electronic symbols (Fig. 2-1) and piping symbols (Fig. 2-2, 2-3). Significantly, however, everyone does not follow the same standard. For example, Ontario Hydro single line symbols are compared with those found in the American Standard examples:

Ontario Hydro **American Standard**

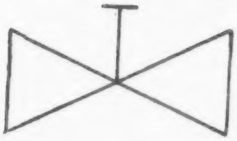


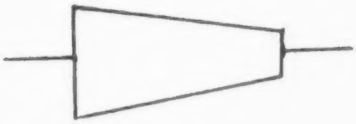


1) Joint Expansion

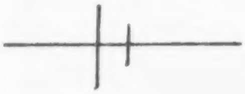


2) Gate Valve





3) Bushing



NOTE: The American Standard Symbols also denote screw type as opposed to flange, bell and spigot, welded or soldered. A symbol can demonstrate what it is and how it is assembled. One should determine just what set of standards are being viewed to fully appreciate what is observed.

The symbol used by Ontario Hydro to represent a Gate Valve is the same as the symbol used in American Standard drawing practice. However, even though this similarity will occur throughout the conventions for single line fittings and valves, beware of the exception. When reading diagrams or drawings constantly refer to the legend which has been provided by the draftsman. In the absence of a legend, and whenever there is room for doubt or confusion, the draftsman will provide a description of the item.

Fig. 2-2 illustrates the single line conventions for fittings and valves.

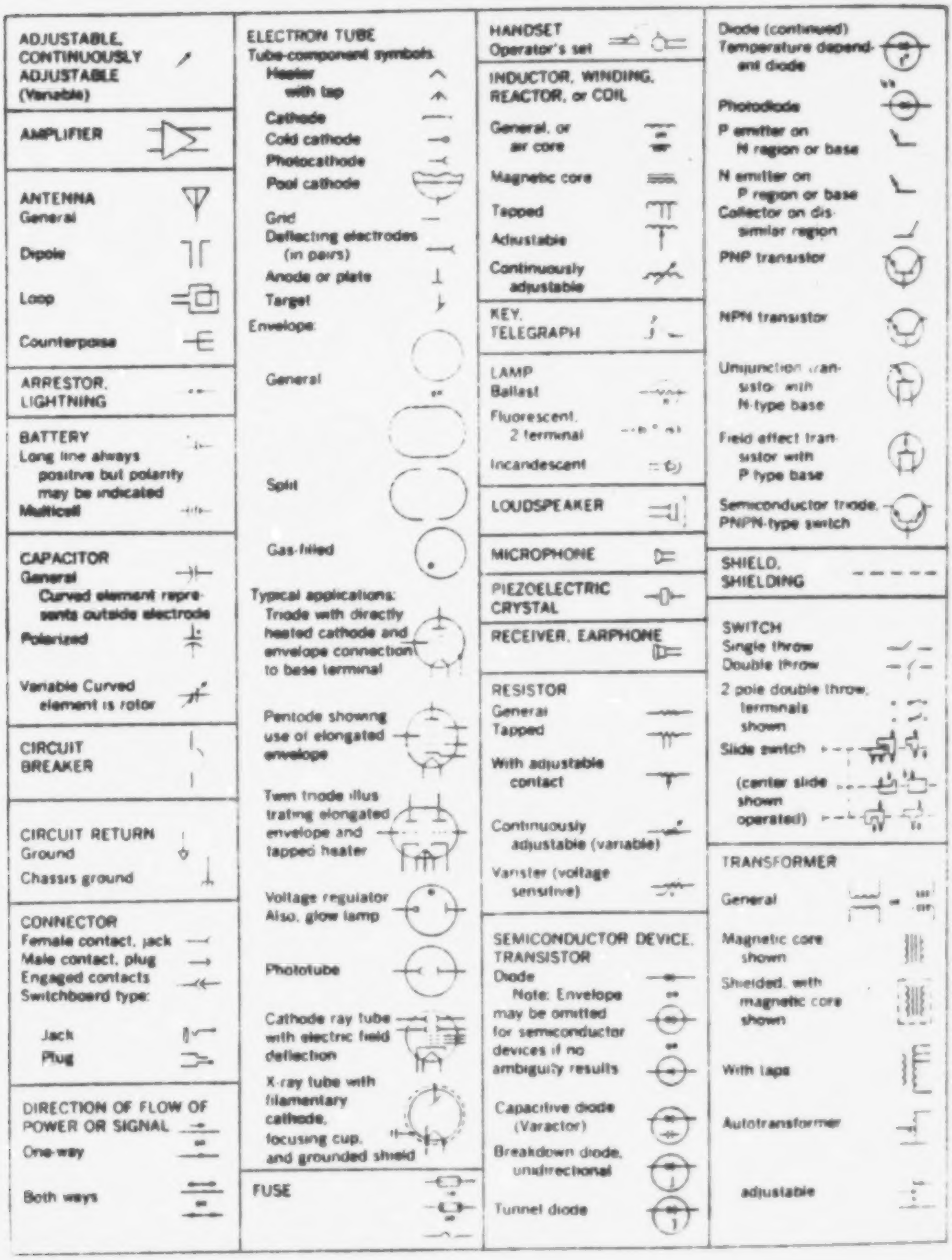
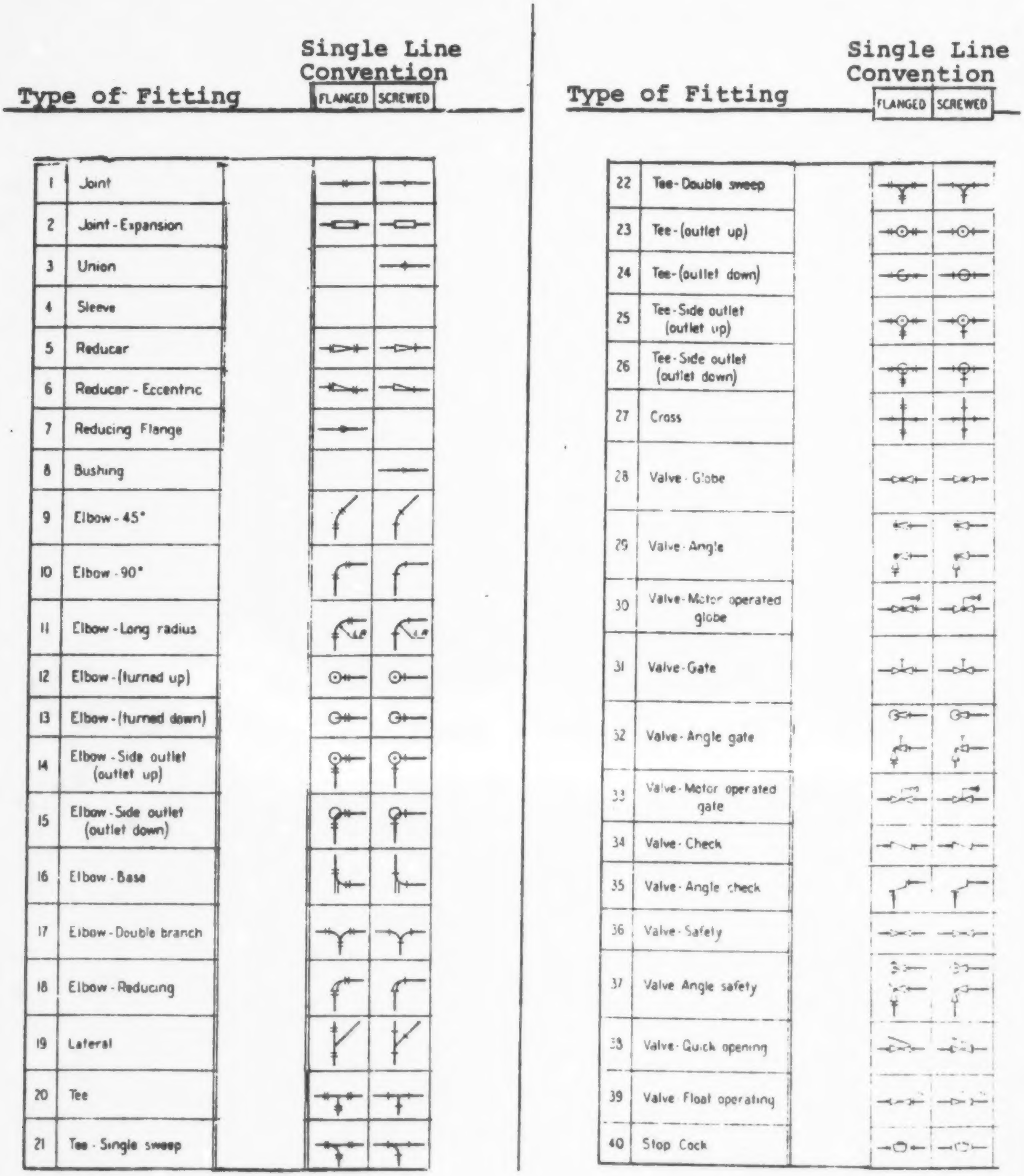

ADJUSTABLE, CONTINUOUSLY ADJUSTABLE (Variable)
AMPLIFIER
ANTENNA
General
Dipole
Loop
Counterpoise
ARRESTOR, LIGHTNING
BATTERY
Long line always positive but polarity may be indicated
Multicell
CAPACITOR
General
Curved element represents outside electrode
Polarized
Variable Curved element is rotor
CIRCUIT BREAKER
CIRCUIT RETURN
Ground
Chassis ground
CONNECTOR
Female contact, jack
Male contact, plug
Engaged contacts
Switchboard type:
Jack
Plug
DIRECTION OF FLOW OF POWER OR SIGNAL
One-way
Both ways
ELECTRON TUBE
Tube-component symbols
Heater
with tap
Cathode
Cold cathode
Photocathode
Pool cathode
Grid
Deflecting electrodes (in pairs)
Anode or plate
Target
Envelope:
General
Split
Gas-filled
Typical applications:
Triode with directly heated cathode and envelope connection to base terminal
Pentode showing use of elongated envelope
Twin triode illustrating elongated envelope and tapped heater
Voltage regulator Also, glow lamp
Phototube
Cathode ray tube with electric field deflection
X-ray tube with filamentary cathode, focusing cup, and grounded shield
FUSE
HANDSET
Operator's set
INDUCTOR, WINDING, REACTOR, or COIL
General, or air core
Magnetic core
Tapped
Adjustable
Continuously adjustable
KEY, TELEGRAPH
LAMP
Ballast
Fluorescent, 2 terminal
Incandescent
LOUDSPEAKER
MICROPHONE
PIEZOELECTRIC CRYSTAL
RECEIVER, EARPHONE
RESISTOR
General
Tapped
With adjustable contact
Continuously adjustable (variable)
Varistor (voltage sensitive)
SEMICONDUCTOR DEVICE, TRANSISTOR
Diode
Note: Envelope may be omitted for semiconductor devices if no ambiguity results
Capacitive diode (Varactor)
Breakdown diode, unidirectional
Tunnel diode
Diode (continued)
Temperature dependent diode
Photodiode
P emitter on N region or base
N emitter on P region or base
Collector on dissimilar region
PNP transistor
NPN transistor
Unijunction transistor with N-type base
Field effect transistor with P type base
Semiconductor triode, PNPN-type switch
SHIELD, SHIELDING
SWITCH
Single throw
Double throw
2 pole double throw, terminals shown
Slide switch
(center slide shown operated)
TRANSFORMER
General
Magnetic core shown
Shielded, with magnetic core shown
With taps
Autotransformer
adjustable

Figure 2-1 American Standard Graphical Symbols for Electronic Diagrams



	Type of Fitting
1	Joint
2	Joint - Expansion
3	Union
4	Sleeve
5	Reducer
6	Reducer - Eccentric
7	Reducing Flange
8	Bushing
9	Elbow - 45°
10	Elbow - 90°
11	Elbow - Long radius
12	Elbow - (turned up)
13	Elbow - (turned down)
14	Elbow - Side outlet (outlet up)
15	Elbow - Side outlet (outlet down)
16	Elbow - Base
17	Elbow - Double branch
18	Elbow - Reducing
19	Lateral
20	Tee
21	Tee - Single sweep
22	Tee - Double sweep
23	Tee - (outlet up)
24	Tee - (outlet down)
25	Tee - Side outlet (outlet up)
26	Tee - Side outlet (outlet down)
27	Cross
28	Valve - Globe
29	Valve - Angle
30	Valve - Motor operated globe
31	Valve - Gate
32	Valve - Angle gate
33	Valve - Motor operated gate
34	Valve - Check
35	Valve - Angle check
36	Valve - Safety
37	Valve Angle safety
38	Valve - Quick opening
39	Valve Float operating
40	Stop Cock

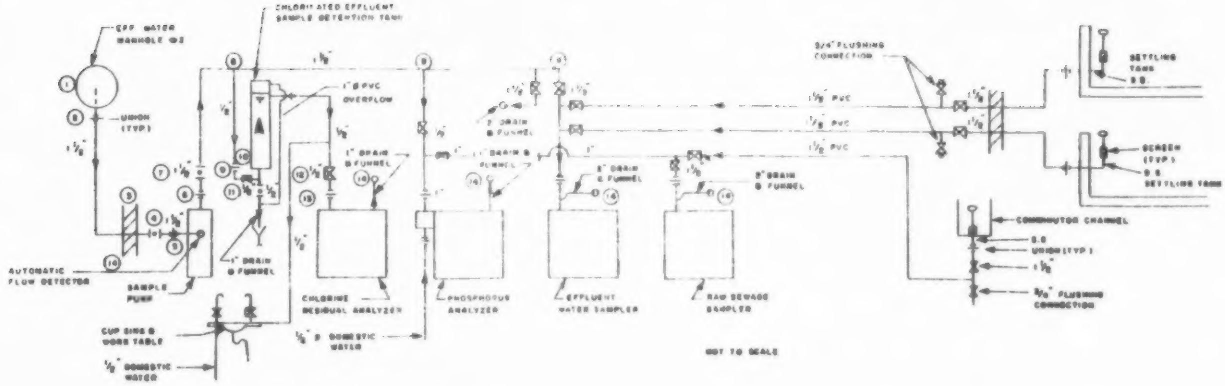
Figure 2-2 Piping Symbols

INTERPRETATION OF SCHEMATICS

This section considers the use of symbols in the schematic drawing. The symbols serve as a means of simplifying the graphic description of a component such as a valve or pump. The schematic drawings simplify the more complex mechanical/ construction drawings of a project, as well as demonstrate the relationship of the components of the system. With this in mind several schematics may be drawn to clarify the major parts of a project. See (Figure 2-3) which is taken from the flow diagram schematic of a small sewage treatment plant.

In the schematic, two items should be noted. Firstly it is not drawn to scale (N.T.S.). This emphasizes the purpose of the schematic: to show the links of the system, not how to construct it in detail. Secondly there are several legends placed directly on the drawings. These provide the definition of the symbols used, and eliminate possible misinterpretation through the use of an incorrect standard. Also provided in the legend are specifics on the item from material type to supplier. This particular schematic covered chlorination, water supply, chemical and sampling systems. For ease of reference during the following discussion of the sampling system, the components in the direction of sample flow are numbered.

NOTE: The sample pump is not identified by symbol but actually stated for what it is. One can see that the sample pump can be isolated by the ball valves (4) and (7) or removed through use of the unions (5) and (6).



LEGEND

SYMBOL	DESCRIPTION	MATERIAL	TYPE CLASS	TYPICAL MANUFACTURER	SERIAL Nº FIG ETC MODEL Nº
———	SAMPLING LINE FITTINGS	PVC PVC	SCHEDULE 80 SOCKET WELD HEAVY DUTY	——— ———	——— ———
—|○|—	BALL VALVE	PVC	TRUE UNION TYPE	CHEMLINE PLASTICS LTD	CTU SERIES
	SAMPLING PUMP	———	PROGRESSING CAVITY 10 GPM	MOYNO	———
—⋈—	SAMPLING LINE VALVE	C.I. NEOPRENE	DIAPHRAGM WEIRLESS TYPE	TRANS SAUNDERS	KB 31-E-T
—⋈▸	FLUSHING CONNECTION HOSE VALVE	BRONZE	WITH CAP & CHAIN	JENKINS	FIG. 303 & FIG. 880
▮	INTAKE SCREEN	STAINLESS STEEL	SEE DETAIL M 10	———	———
	CHLORINE RESIDUAL ANALYZER	———	0-2 MG/LITRE	FISCHER & PORTER	MODEL 17T2 4-20 MA
	PHOSPHORUS ANALYZER	———	DUAL RANGE 0-5 0-0 [illegible]	DELTA SCIENTIFIC	SERIES 9040 4-20 MA
	EFFLUENT SAMPLER	———	FLOW PROPORTIONAL	SIRCO	SERIES B/ST-VS WITH FTC ADAPTOR
	RAW SEWAGE SAMPLER		FLOW PROPORTIONAL	SIRCO	SERIES B/ST-VS WITH FTC ADAPTOR
○	AUTOMATIC FLOW DETECTOR	C.I. RUBBER LINED	FLANGED	MOYNO	MOYNO MONITOR

Figure 2-3 Sampling System Diagram

The versatility of a system can be quickly detected and commented upon for change, if it is required and before it has become fact.

To further simplify the schematic the automatic flow detector (14) is not given a complicated symbol, simply an "0" and this is stated in the legend of the schematic. At this point, (8) the flow can be split into three different uses from the common 1 1/2" line:

1. chlorine residual analyzer
2. phosphorus analyzer
3. sample collection

the diameter of the piping is shown and there is a reduction to one-half inch for each of the above functions.

Numbers 10 and 11 are both valves but two different types. The symbols show what they are and the legend gives specifics, 10 providing flow control, 11 used for straight discharge, not flow regulation. Items such as the detention tank are labelled on the drawing number 12, the sampling line valve provides the ability to control the rate of flow.

One can readily establish that the rate of flow can be adjusted into each of three areas and it can be varied and measured as well.

Using the same schematics, one can identify the components in the lines from settling tank to chlorine and phosphorous analyzers.

What has the schematic done? It has provided a built-in legend to identify the symbols used, and the specifics about each item which requires the additional information. It has provided a simplified but a complete representation of the arrangement of the components of one part of a large project. It therefore gives one the means to assess the capabilities of the system that the operator will be required to maintain.

PLAN READING

Before anyone can interpret plans and drawings correctly he first must have a knowledge of the elements of drawing. Drawing may be defined as the graphic language in which are expressed and recorded the ideas and information necessary for the building of machines and structures, as distinguished from drawing as a fine art.

Because drawing is actually a graphical language, it can only be understood by someone familiar with the language. It cannot be read aloud but must be interpreted by forming a mental picture of the subject concerned. When instruments are used, the language is that of mechanical drawing; when it is accomplished with the unaided hand, it is known as freehand drawing or technical sketching.

Drawing Principles

In drawing any object, full information is required regarding its shape and size; and the problem in engineering drawing is to reproduce the three dimensional shapes of the object, on the surface of a sheet of paper which has only two dimensions. Artists draw objects as they appear and not as they really are. This is "perspective drawing".

To meet the requirement of engineering drawing, the system of "orthographic projection" is used. Practically, this means that the drawing consists of a set of separate views of the object taken by the observer from different positions and arranged relative to each other in a definite way. Each of these views will show two of the three dimensions, and thus a combination of views will show all three dimensions.

To illustrate orthographic projection, the observer may imagine that the object is surrounded by a transparent glass box with hinged sides. The projections on these sides, made by extending perpendiculars from the object, are what would be seen by looking straight at the object from positions directly in front, above and from both sides, as well as from the bottom and rear. These sides of the box, or "planes", are then to be thought of as being opened up into one plane, the plane of the paper. The projection on the front plane is known as the front view or front elevation; that on the horizontal plane, the top view or plan; and that on the side or "profile" plane, the side view, and view or side elevation. Only rarely are the bottom and rear view required. The most usual combination of views is top, front and right side views. Sometimes the left side view is shown if it describes an object more clearly.

From the foregoing study the following principles will be noted:

1. The front view in a drawing indicates the width and height of an object but not the depth from front to back.
2. The top view is directly over the front view and gives the width and depth of the object.
3. The side views are in line horizontally either with the front or top view (usually the front view) and show the height and depth.
4. The depths of the side view are exactly the same as the depth of the top view.

To describe an object completely a drawing should contain lines representing all the edges, intersections and contours of the object. In any view there will be some parts of the object that cannot be seen from the position of the observer as they will be covered by portions of the object closer to the observer's eye. The edges, intersections and contours of these hidden parts, although actually invisible, have their location indicated by the symbol of a lightly drawn line made up of short dashes, commonly called a "dotted line". In drawing, when two lines coincide the nearer one has precedence and the full line always has precedence. In other words, the edges of an object actually visible in any particular view appear as full lines on the drawing, and edges, not visible, take the form of dotted lines, unless coinciding with full lines.

If the object is very simple in its interior construction the hidden lines are not hard to read and understand. Often however, when the interior is complicated or when several different pieces are assembled in one place, an attempt to show the construction on an exterior view would result in a confusing mass of dotted lines, annoying to draw and difficult to read clearly. In such cases one or more of the views is made "in section". A sectional view or section is a representation in which a part of the object is imagined to be cut away and removed so as to expose the interior. The cutting plane is always indicated on the particular view of the drawing to which the sectional view is related. This cutting plane need not be a continuous single plane but may be changed in direction so as to show the construction to best advantage. Material which has been cut by the section plane is indicated by fine diagonal lines called "cross-hatching", uniformly spaced across the cut surface.

Invisible lines beyond the plane of the section should not be drawn unless necessary for a clear description.

In representing objects that are larger than can be drawn to their natural or full size, it is necessary to reduce the size of the drawing in some regular proportion. Every such drawing has this proportion or scale marked on it, which indicates the number of inches on the drawing that would be equivalent to an actual measurement in feet. Drawings of objects have scales of usually 1/3, 1/4 or 1/2 inch equals one foot; whereas scales for maps are often one inch equal to 200 or 300 feet. The scales should always be clearly indicated on drawings.

After the shape of the object has been described the value of the drawing for construction purposes depends upon the dimensioning and notes to tell the size. Complete dimensions are made up of dimension lines, arrowheads, extension lines, leaders, figures and notes. In all cases, sufficient size and location dimensions should be marked so that the essential elements are properly defined.

Reading a Drawing

A drawing is read or interpreted by forming a mental picture of the object represented. In forming this mental picture, one must first gain a general idea of the shape of the object by a rapid survey of all the views given, then select for more careful study the view that best shows the characteristic shape, and by referring back and forth to the adjacent views see what each line represents.

In looking at any view one should always imagine that it is the object itself, not a flat projection of it, that is seen, and in glancing from one view to another the reader should imagine himself as moving around the actual object and looking at it from the direction the view was taken.

The actual size of the object or structure or any part thereof, may be observed by reading the dimensions or by measurement of the drawing itself, using the particular scale involved.

There are two types of drawings that can be most beneficial to the plant operator, the construction drawings and the mechanical drawings.

Construction Drawings

Its purpose is to present the information that is required to build the structure which will hold the equipment needed. The contractor will use these drawings to establish elevations from the footings to the roof tops. These drawings include the detail regarding where to place the reinforcing rods, the type of concrete to use and the size of steel. The larger plants will require many construction drawings in order to show all the detail required. By means of sectioning one view into several subsections, a complex plan can be simplified. In this way the measurements can be more detailed allowing for more accurate installation of equipment.

Mechanical Drawing

Mechanical drawings illustrate the location of non-structural items such as pumps and piping. This drawing will also include the location of future design items, usually shown by a broken line drawing.

By making use of these drawings, one can observe the structural requirements, the location of the equipment and future design considerations. Finally, an overall view is given showing the relationship of the parts of the system, one to the other so that it is possible to form a mental picture of the complete project.

> Study a drawing systematically section by section; structural to mechanical to schematic flow diagrams as required. Generally, no one drawing will show all the information.

SUBJECT:

PREVENTIVE MAINTENANCE

TOPIC: 3

MAINTENANCE OF PIPING AND VALVES

OBJECTIVES

The trainee will be able to:

1. Recall methods of identification of piping.
2. Recall scheduling of pipeline maintenance.
3. Recall the following methods of pipeline cleaning:
    (a) mechanical
    (b) hydraulic
    (c) thermal
    (d) chemical
4. Recall the procedures involved in the cleaning of sludge and scum pipelines.
5. Recall the use and characteristics of valves.
6. Recall the maintenance procedures of valves.
7. Recall the safety factors on pipeline maintenance.

IDENTIFICATION/MAINTENANCE OF PIPELINES

Identification

It is essential that, when a number of pipelines conveying various fluids (liquids, gases and/or slurries) are stacked together or are passed through wall openings into other rooms or passageways that some means of identification be applied to each pipeline. There are several pipeline identification systems available, including the following:

1. Colour Code
2. Descriptive Identification
3. A Combination of 1 and 2

Special markings may also be required to warn operators of the direction of flow, dangerous fluids within the pipe, and dangerous conditions outside of the pipe. Each application must be considered in detail before acceptable procedures and rules can be adopted. The procedures and rules should be reviewed and evaluated on a continuing basis to take advantage of new techniques and equipment, and, in addition, to comply with changes to government regulations.

Colour Coding (See Fig. 3-1)

During the initial stages of treatment plant development, some design engineers and plant operators have displayed a marked preference towards Colour Coding. Unfortunately, the disadvantages of a colour code far outweigh the advantages. Some of the disadvantages of colour coding are as follows:

1. The number of contrasting basic colours is limited and will be found to be inadequate as additional processes are introduced to comply with governmental minimum quality standards.
2. Operators afflicted by colour blindness are put to a great disadvantage when identification is by colour code only.
3. Identification by colour necessitates the use of a code or cipher (Table 3-1). This is an additional complication that can lead to mistaken identification and possibly to accidents that result in damage to property and/or injury to personnel.
4. Both the original cost and maintenance costs are higher for a multicoloured pipe network than for a one colour network. The inventory of paints required to maintain the multicolour system can be enormous, complicating paint touch-up work after maintenance.

PIPE LINES	COLOUR	INERTOL	VALDURA
Raw Sewage	Grey	307	302
Effluent	Grey	307	302
Raw Sludge	Grey	307	302
Drains (Plumbing)	Grey	307	302
Waste Mixed Liquor	Grey	307	302
Waste Activated Sludge	Grey	307	302
Primary Sludge	Grey	307	302
Tank Drainage Lines	Grey	307	302
Sump Pump Discharge	Grey	307	302
Digested Sludge	Black	321	314
Activated Sludge	Black	321	314
Elutriated Sludge	Black	321	314
Sludge Sampling Lines	Black	321	314
Sludge Sampling Lines or	Grey	307	302
Supernatant	Concrete Grey	309	2062
Circulated Sludge	French Grey	332	Tint to match
Transfer Sludge	French Grey	332	Tint to match
Raw Water	Dark Green	323	308
Flushing Water	Dark Green	323	308
Separated Water	Dark Green	323	308
Make-up Water	Dark Green	323	308
Compressed Air (including aeration and instrumentation system)	Light Brown	334	303
Potable or City Water	Blue	301	335
Special Coolant	Royal Blue	303	317
Hot Water	Light Green	336	813
Hot Water Engine Oil Cooling	Light Green	336	813
Heating System Piping	Wall Colour	-	-
Electrical Conduits	Wall Colour	-	-
Sludge or Digester Gas	Red	314	310
Natural Gas	Red (with one band of Buff)	314	310
Lime Slurry	Ivory	321	315
Fuel Oil	Maroon	316	307
Exhaust Piping	Maroon	316	307
Flue from Boilers	Maroon	316	307
Chlorine (liquid)	Yellow (Hansa Type)	319	6309
Chlorine (gas)	Yellow (Hansa Type)	319	6309
Chlorine (Solution)	Yellow (Hansa Type)	319	6309
Ferric Chloride	Light Orange	324	313
Filtrate	Dark Orange	Tint to match	328
Diesel Oil	White	311	318
Lube or Dirty Oil	Tan	325	334
Clean Oil	Buff	313	241
Combustion Air to Engine	Buff	313	241
Vacuum Piping	Buff	313	241
Steam	Aluminum	320	90
Condensate Return	Aluminum	320	90
Vents (relief on boilers)	Aluminum	320	90
Engine Crankcase Breather	Dark Brown	318	329
Cold Water Piping	Sea Green	304	6511
Engine Jacket Water	Misty Green	342	321

Figure 3-1 Colour Code - Pipe Stack

Descriptive Identification

This method of identification is normally applied to a one colour pipe network and, although less pleasing to the eye than multi-colour network, it does have certain advantages when compared to colour coding. They are:

1. Lower original cost
2. Lower maintenance costs
3. Lower paint inventory costs (for touch-up)
4. Eliminates the need for a code or cipher
5. Makes life easier for the colourblind operator
6. Ensures that piping, removed for maintenance, is returned to the pipeline the correct way. This is very important whenever direction of flow indicators (arrows) are employed.

There are two common methods of applying the description. They are by adhesive labels or by stencilling. Of the two, stencilling is possibly the cheapest, and the means to identify equipment is always at hand once the stencilling set is purchased. On the other hand, applying adhesive labels is perhaps less time consuming and is made more attractive by the use of coloured labels. However, it may not always be possible to anticipate future requirements and this may lead to delays in the supply of labels unless a varied stock of labels is maintained at all times.

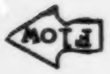
Care must be taken to ensure that whenever pipe covering is removed for maintenance purposes it is either reinstalled or if replaced it is relabelled.

Combination Colour Coding/Description Identification

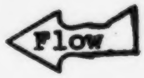
As the title implies, this system of pipeline identification employs both colour coding and description. It is sometimes used as a transitional stage from the colour code system to the descriptive system whenever the size and complexity of the plant processes has outgrown the availability of standard contrasting colours, or when the cost of maintaining a complex colour code is prohibitive. Governmental regulations applied to the identification of pipelines conveying dangerous fluids may prevent a complete change-over to descriptive identification by requiring a colour code. Nevertheless, it is recommended that in such special cases a combination of the required colour code and the description be applied.

Direction of Flow

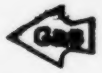
Flow indicators must be applied to the pipeline at regular intervals on straight pipe runs, whenever the pipeline passes through wall, floor and ceiling openings, and whenever the flow may be confusing to the operator by complicated valving and interconnections. Direction of flow indicators must be provided with a means of determining which is the upper side of the arrow, thus:



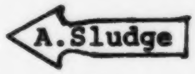
Wrong



Right



Right



Right

The ability to orient the directional arrow is essential to prevent a length of pipe or pipe fitting, removed for maintenance, being inverted when returned to the pipeline and indicating a false direction of flow.

Dangerous Fluids

Steam, hot water, acidic and/or caustic solutions, as well as toxic and/or explosive gases, should be identified either by colour code, description, or a combination of the two. Be generous with the markings, markings should be visible along the pipe run from any position. Valves on pipelines transmitting dangerous substances should also be provided with the capability of being locked in the closed position.

In addition to printed warnings of dangerous contents, we must provide some protective warnings for the sensitive interiors of some pipelines. Pipelines manufactured from non-metallic materials are replacing iron and steel pipe in many treatment processes, and even when metallic pipe is in use it may be lined with either glass, rubber or plastic. Therefore, extreme care must be taken to identify a pipeline that requires special cleaning procedures and tools, otherwise the pipe or lining may be damaged, resulting in expensive repairs or replacement.

Magnetic flow meters employ an insulated tube that is easily damaged by the passage of metal rods and/or augers. A printed warning must be posted on the pipeline at each point of access, together with the approved method of cleaning the pipeline.

Dangerous Conditions Outside the Pipe

Warnings should be posted along the pipeline for the protection of personnel against hidden hazards, i.e. electrical, steam and/or hot water tracing buried within or covered by the insulated covering (lagging) of the pipeline. In the case of electrical tracing it may be necessary to provide an electrical disconnect switch with provision for locking out to conform to safety regulations.

MAINTENANCE

General

Pipelines are, possibly, the most neglected equipment employed in both water and sewage treatment processes, yet they are of prime importance. Rarely are they included in preventive maintenance programs but they are ignored until they break or become plugged. In both cases, water and sewage, the end result is the same, interruption of plant processes for unscheduled maintenance. A relatively small leak from a water main will, if neglected, increase to proportions that may undermine the pavement and/or adjacent pipelines or sewers, and, in addition it may reduce the system head sufficient to cause the overloading of motors on centrifugal pumps. Loss of pressure in the pipeline can result in the infiltration of polluted ground water into the system. Plugged or restricted sludge and/or scum pipelines, on the other hand, will result in an increased system head that can cause the overloading of positive displacement pump motors, failure of shear pins or the operation of other protective devices.

Scheduling of Maintenance

The following procedures carried out routinely can be effective as a means of determining the internal condition of a pipeline without the need for an internal inspection:

1. Observe and record changes in the system head at pressure gauges down stream of the pump.
2. Observe and record changes in flow rates indicated by metering devices.
3. Observe and record changes in motor loading indicated by the ammeter.

The effectiveness of these methods can be improved by the installation of indicator flags to gauges and meters indicating the normal current, pressure, and flow rate.

The chronological scheduling of sludge and scum pipeline maintenance has little to recommend it. Rarely do pipelines plug to a timetable. Whenever the incidence of plugging reaches unacceptable levels the pipeline should be cleaned out back to the sump or hopper supplying the pipeline.

Pipeline maintenance should of course include the pipe exterior elements, such as:

1. Repair and painting of pipe insulation
2. Secure, repair, replace, adjust and paint pipe hangers, brackets, clamps, etc.
3. Tighten flanges and replace gaskets
4. Maintenance of pipeline identification markings

External maintenance of pipelines can be scheduled by hours of operation or elapsed time.

Due to the composition of the material handled, certain pipe-lines are subject to plugging. Included are:

1. Drains from sedimentation and settling basins in Water Purification Plants.
2. Sludge and scum pipelines in Sewage Treatment Plants.
3. Ash slurry pipelines from incineration process in both Sewage Treatment and Solid Waste Disposal plants.
4. Floor drains whenever the protective gratings are removed.

Preventive action plays an important role in the struggle to prevent plugged pipelines. The following are a few examples:

1. Remove all unnecessary restrictions from the sludge pipeline, ie. plug valves with rectangular or diamond shaped ports, diaphragm valves, globe valves, all valves with parts of less than full pipe size openings.

2. Flow meters employing either a Venturi tube or an orifice plate are not recommended for use on sludge or scum pipelines.
3. Magnetic flow meters are recommended for such duty, provided precautions are taken to avoid damage to the insulated tube by the display of warning notices.

4. Long radius elbows and sweep fitting on tees and on cross branches are recommended whenever rods and augers, etc. are used for pipeline cleaning purposes.
5. The plugging of scum pipelines is usually the result of scum being left in the pipeline and allowed to harden. Flushing the pipeline with hot water after each scum pumping session will prevent this type of plugging.
6. In addition to 5 above, whenever the scum is heated the pipeline should be insulated to retard heat loss.
7. Tracing scum pipelines by means of steam, hot water or an electric element is another effective preventive measure. In addition, tracing can be effectively employed in the clearing of blocked scum pipelines. A more common used method is to attach the tracing line to the outside wall of the scum pipeline and then encase both in an insulated covering.
8. The emergency tracing of scum pipelines by passing a steam or hot water tracing through the interior of the scum pipeline has been adopted in the past. It is however, a desperation measure necessitated by the scum line being buried and therefore inaccessible to the operator. This method is not recommended, it has too many disadvantages, such as:

(a) Creating an increase in the system head

(b) Requires troublesome glands where the tracing enters and leaves the scum pipeline

(c) Prevents rodding operations unless removed

(d) It may, in itself, be the cause of mechanical plugging

Characteristics of Steel/Wrought Iron Pipe

Attached for information as Appendix A are notes on the characteristics of steel and wrought iron pipe.

Methods of Pipe Manufacture

Attached for information as Appendix B are notes on the methods of pipe manufacture.

PIPELINE CLEANING

There are at least four methods of cleaning pipelines:

1. Mechanical (rods, hammers, augers and cutters, etc.)
2. Hydraulic (high pressure hoses, high velocity nozzles)
3. Thermal (steam, hot water and electric element)
4. Chemical (acids or caustic solutions)

Mechanical Cleaning

Pipeline cleaning equipment is available in such a wide variety and from so many reputable suppliers that it would not be fair to discuss any particular type or manufacturer in detail. With minor differences they are all very much alike at first glance, it is only after using equipment for some time that a true evaluation can be made. As much information as possible should be obtained before ordering equipment. A reputable supplier will gladly provide the names and phone numbers of their customers, from whom information can be obtained.

Hydraulic Cleaning

Hydraulic cleaning can mean everything from a flushing procedure to high pressure cleaning with high velocity nozzles. High pressure hoses can be used with turbine driven cutters and hammers to remove hard scale and deposits as for instance are found in boiler tubes, etc. High velocity nozzles are effective in removing semi-hard, clay or sandy deposits by the cutting action of the high velocity jets of water. Hydraulic cleaning can of course be allied to mechanical cleaning to flush out the loosened solids.

Thermal Cleaning

Thermal cleaning is usually applied to grease and scum build up in pipelines, it can be effectively used as a preventive measure as with tracing. Portable steam generators are extremely useful for the direct removal of grease. They are preferable to fixed boiler installations where the boiler feed water is treated and the unrestricted use of make-up water is to be discouraged. In addition, a fixed boiler system may require a prodigious amount of piping to cover the entire plant.

Chemical Cleaning

Both acidic and caustic solutions are involved in the chemical cleaning of pipelines, however, it may be somewhat dangerous to generalize on the subject of chemical cleaning. The use of chemicals should not be undertaken unless professional opinion is obtained and followed, at least initially and whenever a change of chemical or procedure is anticipated. Extreme care must be taken and protective clothing and equipment worn whenever chemicals of this nature are being used.

A job is never completed until the tools have been returned to their proper place, and the job site has been cleaned up and is restored to an acceptable level of cleanliness.

PIPELINE CLEANING (SLUDGE & SCUM)

Preventive measures notwithstanding, pipelines will, for various reasons, become restricted or plugged. It is essential that timely and adequate preparations be made to effectively deal with the problem when it does occur. Preparations should include the following:

1. Divide the pipeline into sections of approximately 150 feet, or in lengths within the range of the cleaning equipment.
2. Install a means of access at each end of each section of pipeline. Whenever the direction of the pipeline is changed at an access point install either a tee or a cross branch. When a section terminates in a straight run of pipe install a short makeup piece. Be careful to avoid creating excessive system head. Remember that a 6" Tee in branch is equivalent in resistance to flow, to 15 feet of cast iron pipe.
3. To speed up the opening and closing process of the pipeline it is recommended that quick operating pipe connections be employed, similar to either Victaulic 2-bolt or Victaulic snap-joint couplings.

The practice of using 4 bolts to secure flanges drilled for an 8 bolt fastening should be discouraged. Not only is it unsightly and hard on flange gaskets, it is also a dangerous and lazy habit that may be carried over to high pressure applications.

4. Flanges and pipe fittings at access points should not, for obvious reasons, be covered with insulated pipe covering.

5. Pipe hangers are not suitable for supporting pipelines that will require cleaning by rods, augers, etc. The pipeline should be rigidly clamped to wall brackets at a height of approximately 3 to 4 feet, so that effective rodding operations can be carried out (Fig. 3-3).

6. Whenever pipelines are to be stacked in two or more vertical rows, pipelines that will require cleaning should be located to the inside of the stack, that is, nearest to the operator.

7. It is recommended that each access point be provided with adequate lighting, a power outlet, a supply of flushing water and adequate drainage. In addition, it is recommended that the floor at each access point be coated with a non-slip paint.

8. Whenever a pipeline is manufactured of material other than iron or the pipeline is lined with rubber, plastic, glass or any substance not compatible with the established cleaning methods, a warning notice must be posted at each access point.

9. Personnel must also be warned not to pass steel cleaning rods and/or augers, etc. through the tube of a magnetic flow meter, otherwise the tube will sustain damage that will be costly to repair.

USE AND CHARACTERISTICS OF VALVES

Line valves are the means by which flow through a pipeline is either started, regulated or stopped. It is essential, therefore, that operators and maintenance personnel be sufficiently knowledgeable to be able to select the correct valve for a given application, then install and maintain it so that it will perform satisfactorily.

Gate Valves

There are two types of gate valves, one with a rising spindle (Figure 3-3) and one with a non-rising spindle (Figure 3-4) used for fully open/fully closed (starting/stopping) flow operations. Some important characteristics are:

1. Tapered gate and seat will cause turbulence and vibrations when only slightly opened. Unless the gate is returned to the seat, damage to both faces can result. Under high head/flow conditions slight unseating such as this will result in "wire drawing" and erosion of both seat faces.

2. High head conditions being applied to one side of the gate could make opening extremely difficult if not impossible. Flooding the dewatered side of the gate to equalize the pressure by means of a by-pass will not only simplify the opening process but will also prevent damage to the seats and/or the spindle.
3. The thread in both the nut and spindle of the valve with the non-rising spindle are exposed to the flow and are not recommended for service when the fluid is contaminated as in sewage treatment. The rising spindle valve type should be used for contaminated fluids.
4. The straight through flow characteristics of the gate valve when fully open ensures a minimum loss of head through the valve.
5. Contaminants in contact with the thread of nut and spindle in the non-rising spindle gate valve will cause seizure, unless the valve is periodically operated.

Note that a multiple thread is often employed to speed up the operating speed (closing/opening) of the gate.

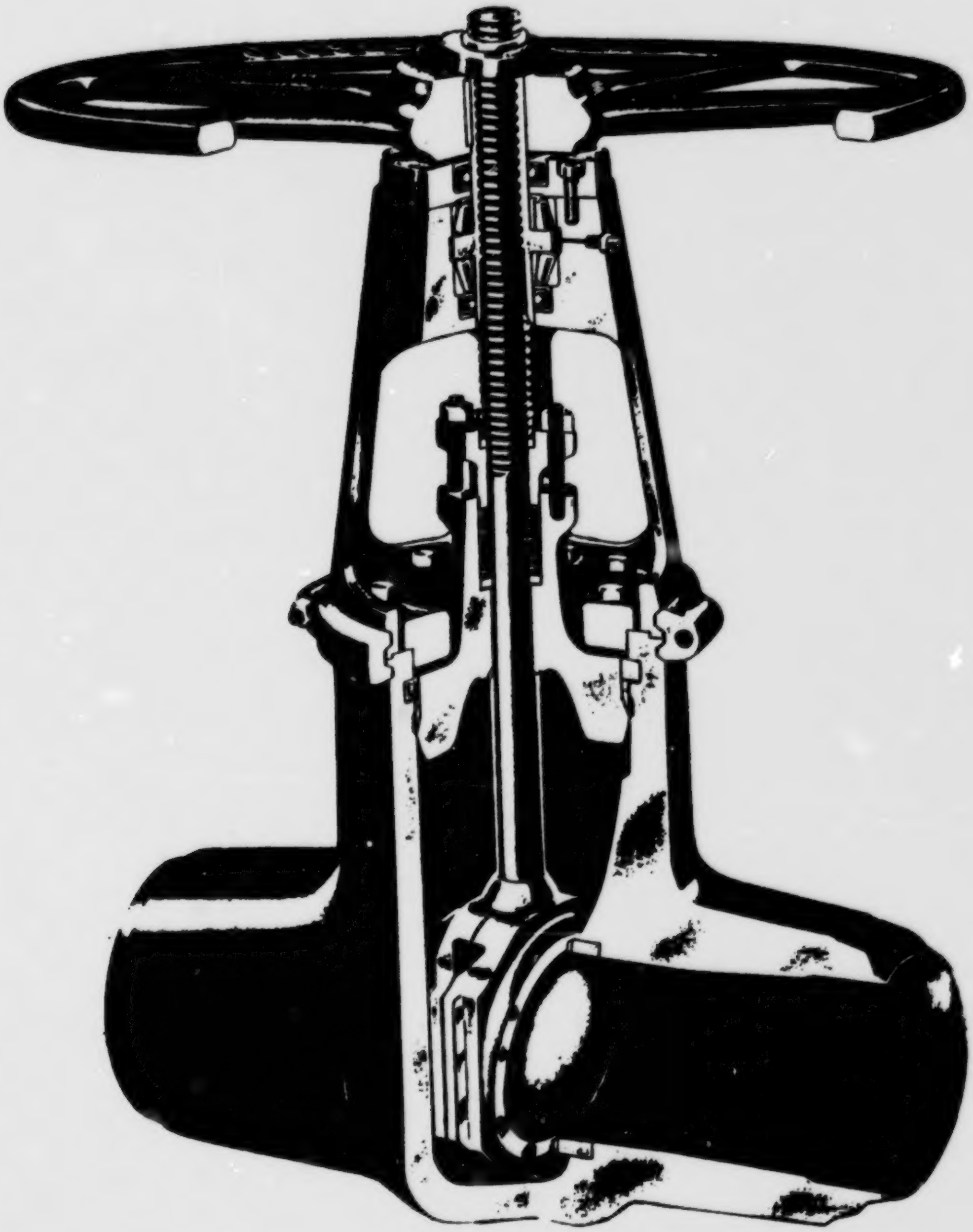


Figure 3-3 Gate Valve Rising Stem

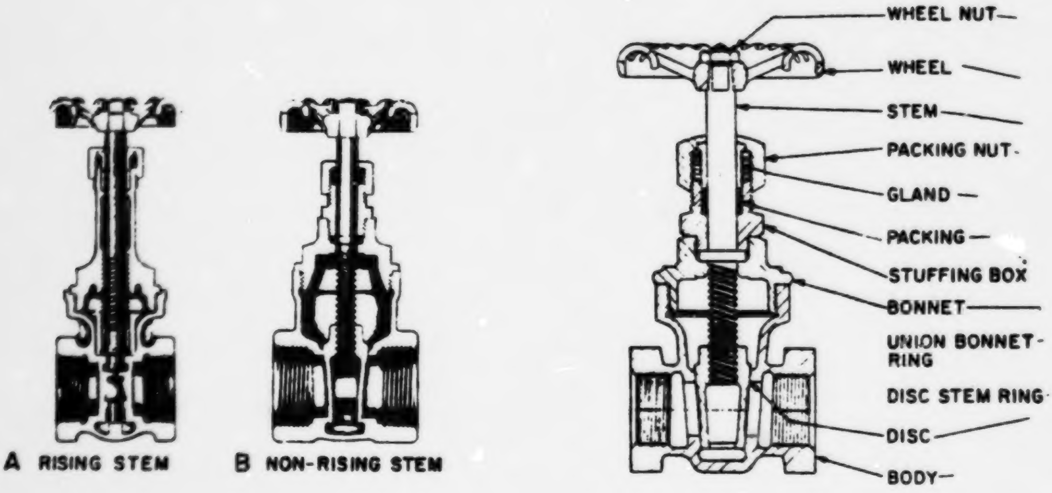


Figure 3-4 Gate Valve - Comparison of Rising Stem and Non-Rising Stem

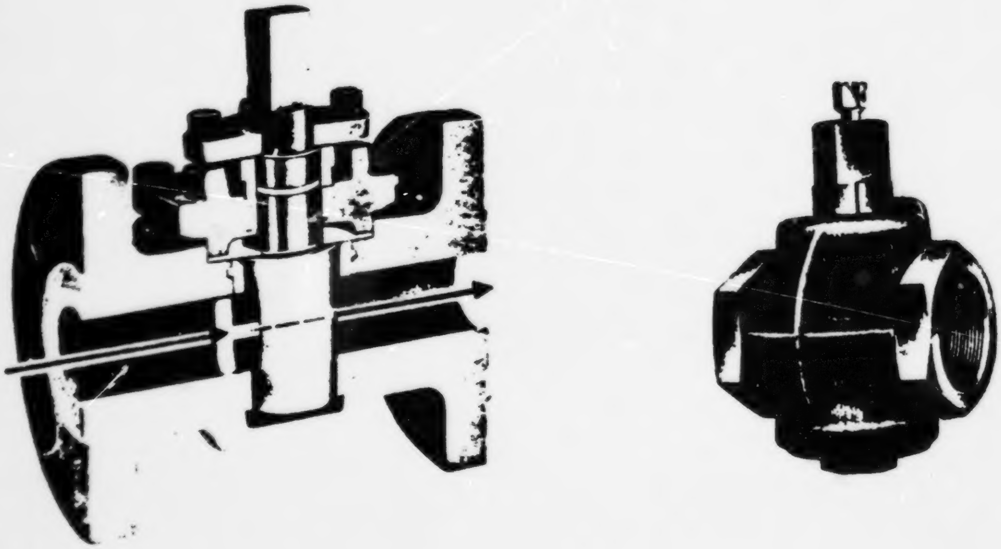


Figure 3-5 Plug Valve, Cutaway and Normal

Plug Valves

There are four types of plug valves, they are:

1. Taper Plug Valve
2. Parallel Plug Valve
3. Eccentric Plug Valve
4. Ball valve

Plug valves are suitable for either partial or fully open/closed operation.

To prevent water hammer when employed on high head applications the speed of operation opening/closing, should be slowed down by suitable gearing.

Characteristics of these types of plug valves which should be noted are:

1. Basically there are only four parts to the plug valve, they are as follows:
   a. The body
   b. The plug
   c. The washer
   d. The nut
2. The taper plug valve is not recommended for service with contaminated fluids such as sewage or slurries. The taper plug requires unseating before it can be turned which may allow particulate matter (grit) to enter the space between plug and seat. The subsequent turning and seating motion may result in scoring the seat and plug.

3. The parallel plug valve has the disadvantage that it requires sealing by an application of compound applied from the outside. The effect of combining grit and sealing compound can well be imagined. This factor alone is more than sufficient reason for excluding the valve for service with contaminated fluids.

4. The eccentric plug valve is an attempt to overcome the deficiencies of the tapered plug valve and the parallel plug valve. The design is compatible with rubber/neoprene coating and lining for use with a corrosive fluid. It is recommended for service in most areas of the sewage treatment process. See cut-away model.

5. Plug valves (Fig. 3-5) can be supplied with round, rectangular and diamond shaped ports. Round port valves are recommended for service where the valve opening must be equal to the diameter of the pipe, i.e. sludge and slurry pipelines. Plug valves with a rectangular port are recommended for general open/shut applications with clear mediums. Plug valves with a diamond shaped port are recommended for flow regulating service.

6. Another variation in plug valve design is the ball valve (Fig. 3-6). As the name implies a ball or sphere replaces the tapered or cylindrical plug. The ball or spherical valve is recommended for flow regulation service. In some high head, high flow, water pumping applications they are displacing the plug valve mainly because of improved performance, reliability and ease of maintenance.

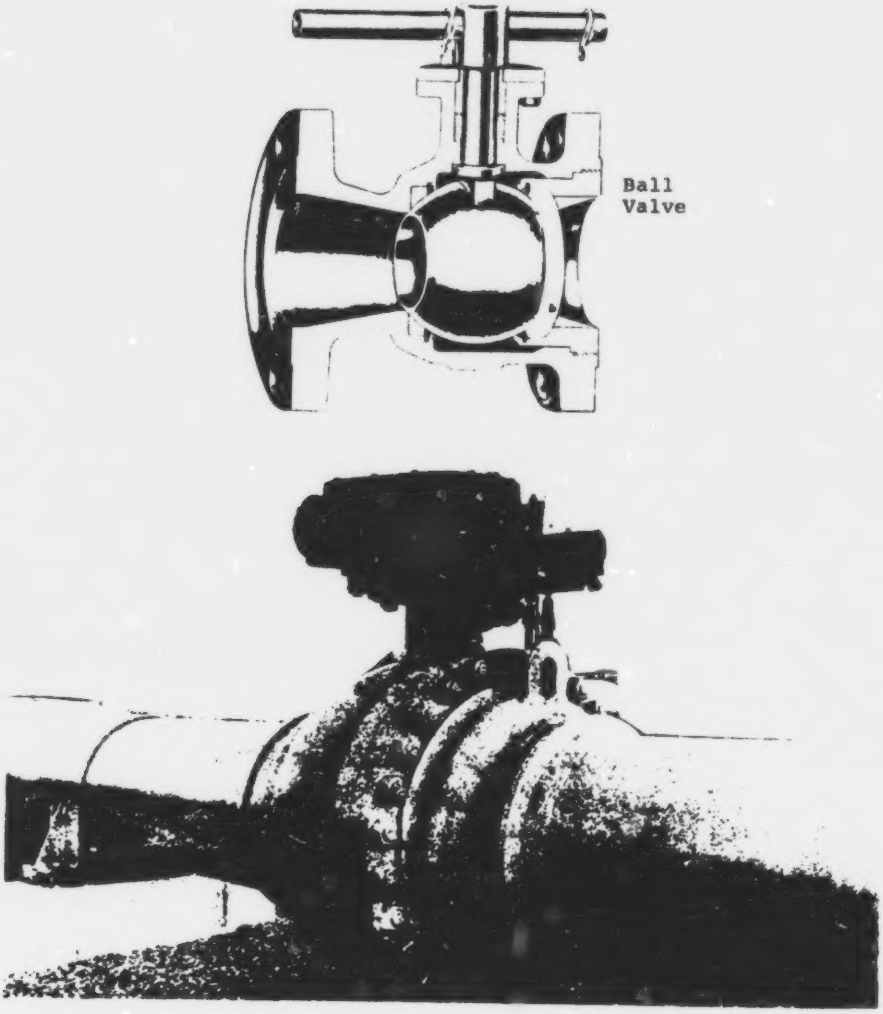


Figure 3-6 Ball Valve Cutaway and Large Installation, Mechanically Controlled

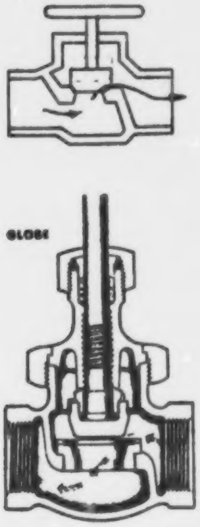


Figure 3-7
General Purpose Globe Valve

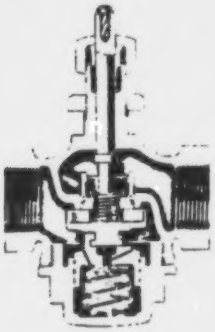


Figure 3-8
Quick Operating Globe Valve
(for use where rapid closure

is

essential).

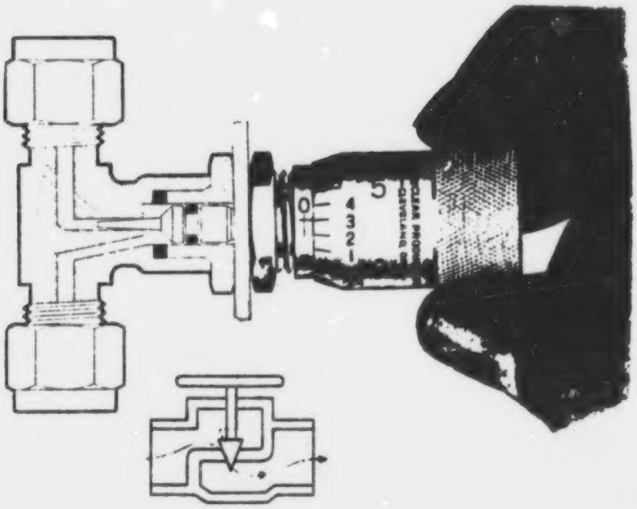


Figure 3-9 Needle Valve

Globe Valve

The globe valve (Figure 3-7) is employed in the most common of all flow regulation applications, the faucet. It is extremely rugged and reliable and will perform for years with little or no maintenance but plenty of abuse. However, it does cause consider-able head loss in a piping system due to the path the fluid must take to pass through the valve. Care must be taken to install the valve in the line so that when the valve is fully closed the pressure is contained under the seat and is not against the stem packing. Some spindles have a back seating disc that will, when the valve is fully open, protect the stem packing from the line pressure.

Needle Valve

The needle valve (Figure 3-9) is similar to the globe valve in that it is recommended for flow regulating service but will provide greater accuracy. Inspection of Figures 3-8 and 3-9 will expose the similarities and differences between the needle and globe valve.

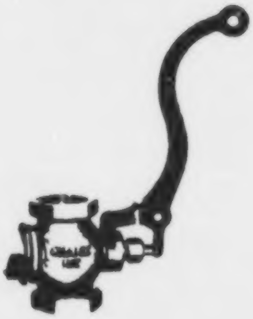


Figure 3-10
Quick Opening Globe Valve

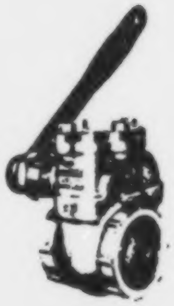


Figure 3-11
Quick Opening Coin Action Gate Valve

Quick-Operating Valves

The globe valve and the gate valve can be easily adapted to quick-opening/quick-closing operation. A lever is used to operate the valve in place of the usual threaded spindle and control wheel (Figures 3-10 and 3-11). The plug valve and the butterfly valve can be used on quick-operating applications when an operating lever is directly connected to the valve spindle.

Air Horns, Emergency Showers, Emergency Eye Washers, and Emergency Gas line shutoffs are but a few examples of applications for quick-operating valves.

Butterfly Valves

Butterfly valves are manufactured for pressures of up to 150 P.S.I. and temperatures of from -40°F to 300°F dependent upon the design and upon the materials used in construction. It is recommended for both flow regulation and for fully open/closed operation. However, some users may challenge claims to suitability for high head applications made by certain manufacturers.

Butterfly valves can be purchased with the spindle in either a vertical or horizontal plane (Figure 3-12) will illustrate a horizontal butterfly valve.

Diaphragm Valves

Diaphragm valves are recommended for applications in which the fluid handled would be harmful to the operating mechanism. It is also suitable for corrosive service because it adapts readily to lining with a wide variety of corrosive and abrasive resistant materials. It does not require a gland and provided the diaphragm remains intact there will be no leakage between spindle and body.

The most common type of diaphragm valve operates on the principal of pinch; the elastomer diaphragm being pinched against a weir cast into the body of the valve (Figure 3-13). A diaphragm can be used in conjunction with globe and needle valves whenever the flow of dangerous fluids must be contained within the valve body.

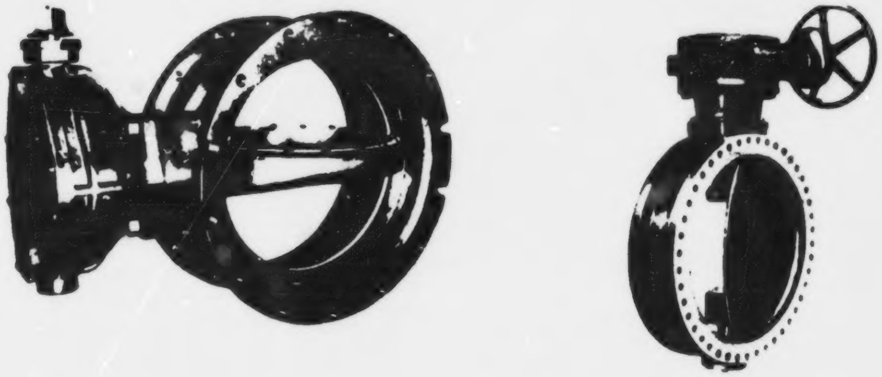


Figure 3-12 Butterfly Valve - Vertical and Horizontal

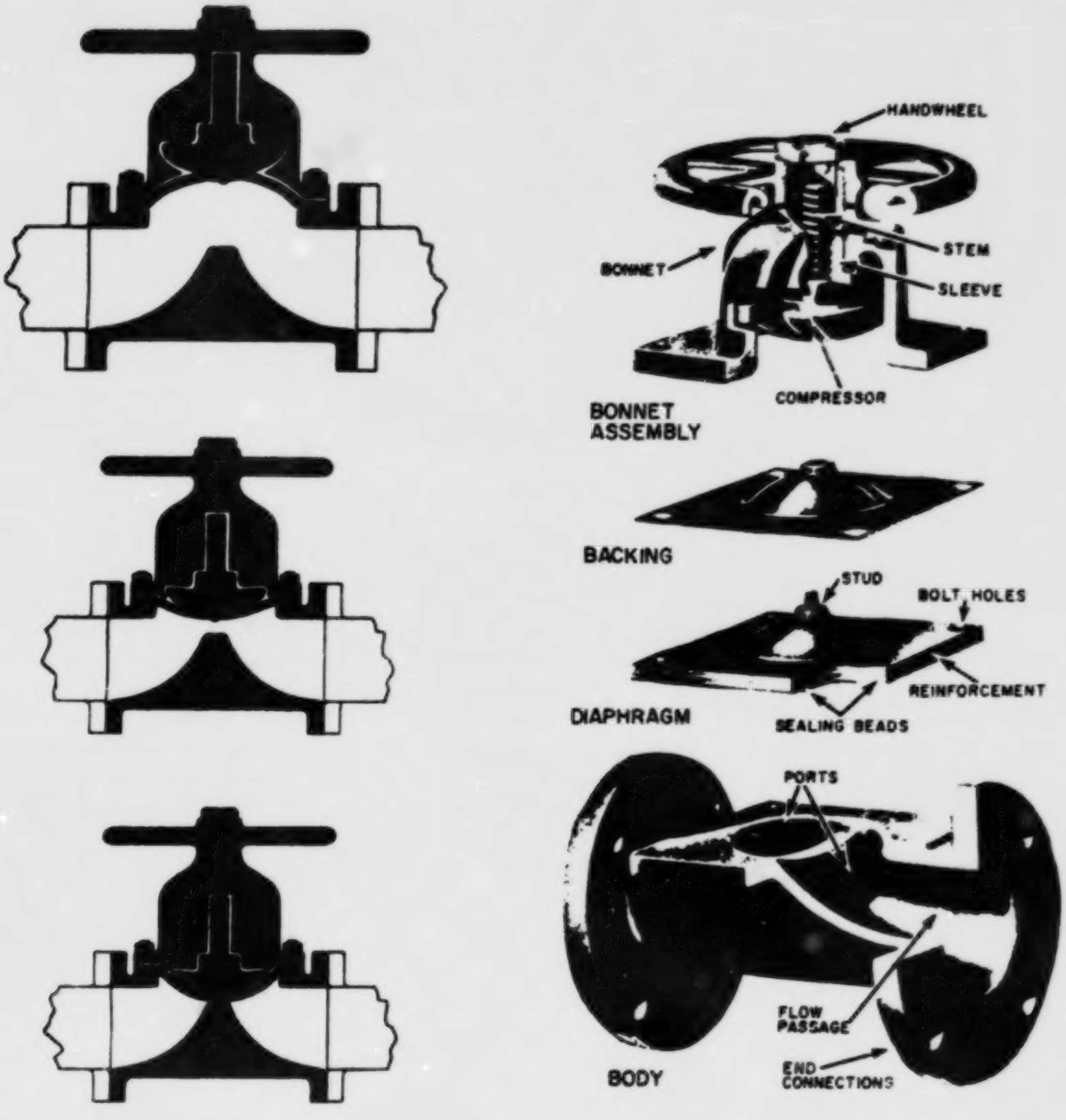
Check Valves

Provision for the control of direction of flow is made by means of check valves. These include:

1. Swing Check Valve Horizontal Line (Figure 3-14)
2. Horizontal Lift Check Valve (Figure 3-15)
3. Vertical Lift Check Valve (Figure 3-16)
4. Ball Check Valve for Vertical pipeline (Figure 3-17)
5. Foot Valve installed at open end of suction wipe to prevent drainage and loss of prime (Figure 3-18)

The cross-sectional view of a swing check valve (Figure 3-14) shows that the valve is connected to the upper part of the body by a hinge. The valve swings about the hinge in response to changes in rate of flow. Whenever the flow stops and attempts to reverse direction the valve will swing into the closed position against the seat located in the body.

Whenever the swing check valve is employed in a high head/flow applications it is recommended that a "Dash Pot" be installed to prevent damage to the valve by reducing the effects of water hammer. The swing check valve has the advantage of a low loss of head through the valve, particularly under full flow conditions. It can also operate in a vertical and a horizontal position.



A

B

Figure 3-13 **Diaphragm Valve**

A - Shows Various Positions
B - Shows Blow-up of Valve

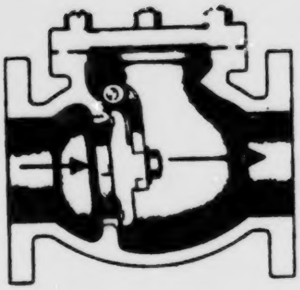


Figure 3-14
Swing Check Valve

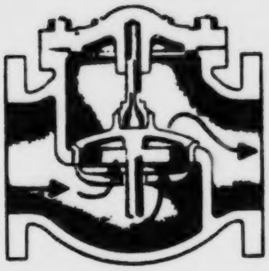


Figure 3-15
Horizontal Lift Check Valve

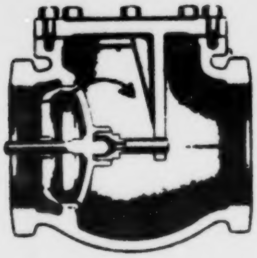


Figure 3-16
Vertical Lift Check Valve

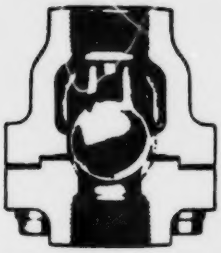


Figure 3-17
Ball Check Valve

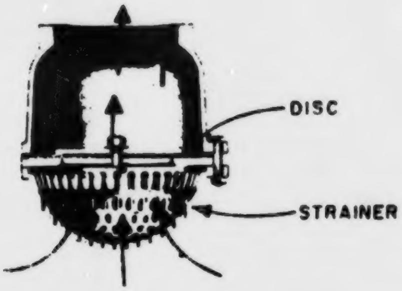


Figure 3-18
Foot Valve

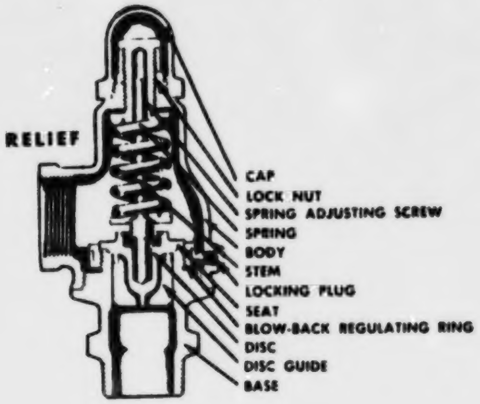
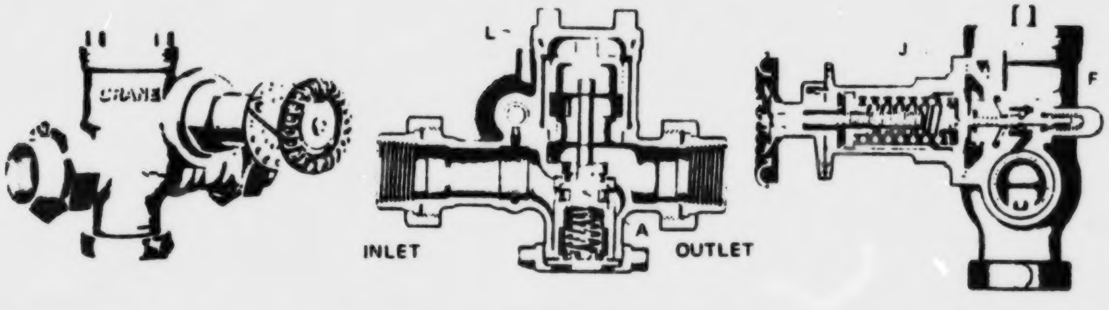


Figure 3-19 Pressure Relief Valve (Safety Valves)



VALVE

MAIN VALVE - AND - PISTON ASSEMBLY

AUXILIARY VALVE ASSEMBLY

Figure 3-20 Pressure Regulating Valve

Special Service Valves

Special service valves are, for the most part, simply common line valves that have been adapted to meet a special application, e.g.:

1. Pressure Relief Valve (Safety Valve) is, a globe valve with a plain, spring loaded spindle, preset to open and close in response to a pressure change (Figure 3-19).
2. Pressure Regulating Valve can also be a globe valve with a plain spindle (unthreaded) capable of modulating the downstream pressure in response to variations to down- stream and/or upstream pressures (Figure 3-20).

Operation of Valves

Line valves are operated by hand (manually), by electric motor or solenoid, by hydraulic or pneumatic powered mechanisms. Certain valves are spring operated but usually in one direction only - closing. The drive can be direct acting as with hydraulic, pneumatic cylinders or diaphragms or by means of an electric solenoid, or indirectly by use of a train of gears interposed between the valve and the driver. Control can be either local or remote, manual, or, automatically in response to signals from a monitoring source.

Method of Connection to Pipeline

Valves are connected to a pipeline by the same methods employed to connect any other pipe fitting (i.e. elbow, tee, cross branch) by screwed connection, flanged connection, compression fitting, Victaulic, soldered joint or welding.

In the case of welding, nipples and flanges should be welded to the valve to make a flanged connection to the pipe-line to facilitate removal. Always bear in mind that sooner or later a valve will have to be maintained so make provision for its removal. Provide proper support for the pipeline adjacent to the valve. Heavy valves of 8 inches or over should be directly supported.

Gate valves located on the suction side of centrifugal pumps should, whenever room permits, be turned so that the spindle is parallel to the ground. This will ensure that the medium being handled will keep the stem packing moist and prevent entrance of air into the pump.

The valve must be free of stress, therefore the pipeline must be properly aligned and the flange faces square. Flanges are to be tightened evenly by the proper rotation of the wrench on the flange bolts (Figure 3-21).

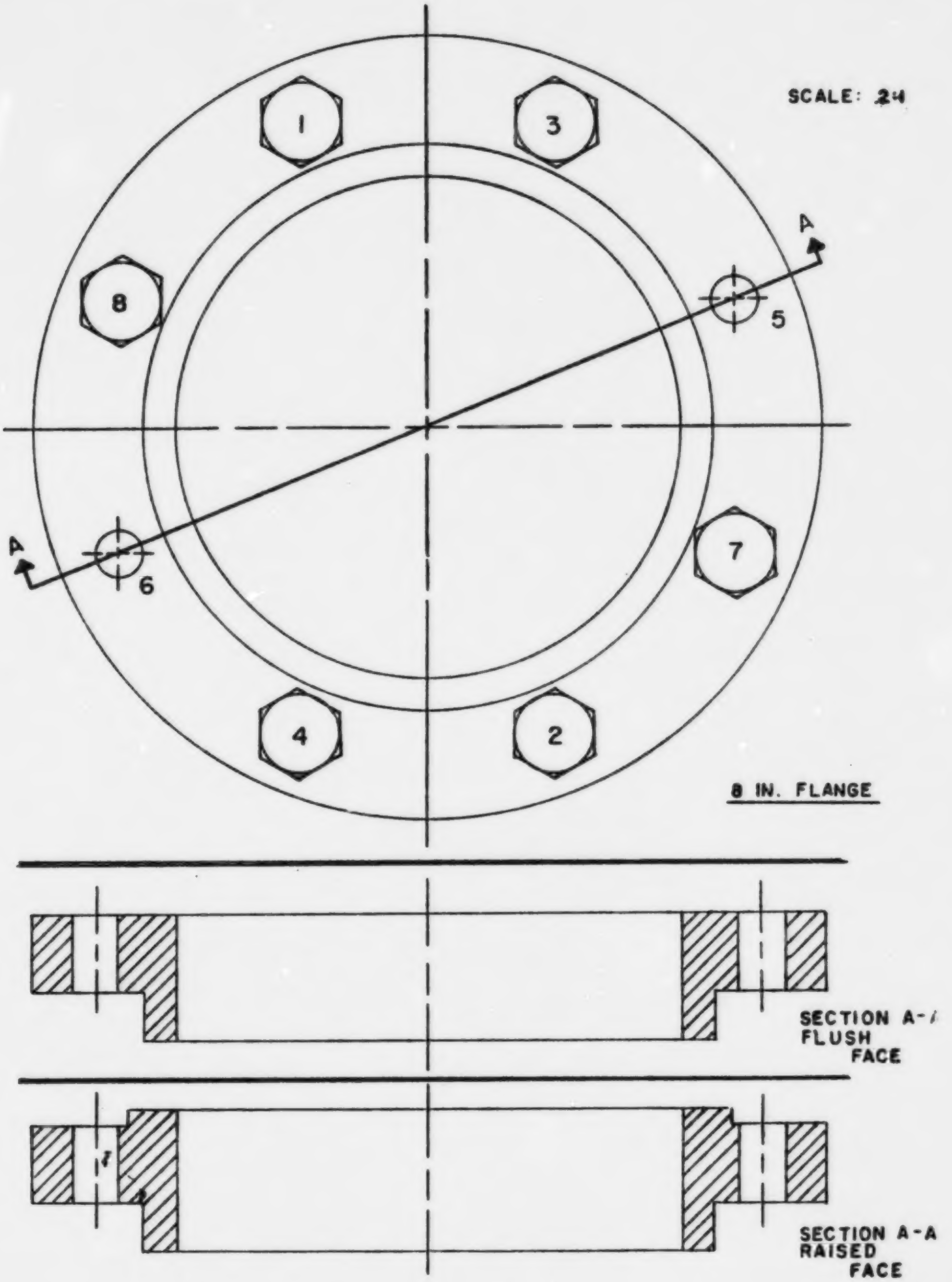


Figure 3-21 Order of Tightening Flange Bolts

IDENTIFICATION/MAINTENANCE OF VALVES

Identification

A single equipment identification system should be capable of accommodating all types of equipment within the area of responsibility. The simplest method of identification is to use a single number or letter, i.e. 1, 2, 3 or a, b, c. This would be adequate provided that identification is limited to a few items of a single type and located in one area or employed on one process, and provided that there would be no future expansion of plant or process.

Design Engineers and Contractors quite often adopt the initials of the item plus a number as a means of identification, i.e. PV = Plug Valve; CV = Check Valve; GV = Gate Valve; GlV = Globe Valve, etc. The alphabetical identifier is suffixed by a numeral to indicate sequence. This system may be adequate for contract work, but it is not adequate for plant operations where the identifier may be required to identify any or all of the following:

1. Location
2. Process
3. Type of equipment
4. Sequence

Thus a simple system might for example use PV #1, where P = location or process (Primary); V = type of equipment (Valve); and #1 = sequence number. There are many variations of this type of code. It is simply a matter of selecting a code that, will satisfy the present and future needs of the plant.

The cost of designing and implementing an identification system can be considerable, therefore the task should not be undertaken lightly or half-heartedly. The identification code should also be applied to the equipment file, drawings, manuals, history of repairs, etc. Whenever an identification code is adopted the identifier must be applied to all components of the equipment by means of a stencil decal or engraved plate. The markings must be clear and permanent and conform to the requirements of the "The Occupational Health and Safety Act 1978 and Regulations for Industrial Establishments".

Maintenance

The descriptions of the valves in the foregoing pages and the various applications for which they are recommended should have provided an insight into the maintenance requirements of valves. Prescribing maintenance requirements without a specific valve in mind, and without knowing what the manufacturer's recommendations are, or the conditions under which the valve operates, etc., can be a risky proposition. Nevertheless, there are certain aspects of preventive maintenance that can be applied to valves in general.

A few words of caution may be necessary before the question of preventive maintenance is discussed. There is a tendency for personnel, when setting up a preventive maintenance schedule, to play it safe by being overgenerous, and allow too short an interval of time between activities. Also, to schedule an unnecessary amount of work at each activity. The question to ask is, not what maintenance procedure is desirable, but rather what maintenance procedure is essential? If the preventive maintenance program is overloaded with non-essentials it will most likely fail.

Part of the routine preventive maintenance effort should be aimed at preventing the valve from seizure by the scheduled periodic opening and closing of each valve. The time interval between servicing will be dependant on several factors:

1. The condition of the fluid
2. Climatic conditions (if outdoors)
3. The ambient conditions (wet, dry, not cold and dusty)
4. The frequency and severity of service

For various valves and their parts see Figure 22.

1. **Gate Valves:** Oil spindle at gland
   Check gland leakage
   Check flange leakage

2. **Plug Valves:**

   (a) Parallel Plug:

   Replenish the sealing compound as required
   Check gland for leakage
   Check flange for leakage

   (b) Taper Plug:

   Replenish the sealing compound or lubricant

   (c) Ball or Spherical Plug

   Check gland for leakage
   Check flange for leakage

3. **Globe Valve:**

   Check gland for leakage
   Check flange for leakage
   Oil spindle at gland

4. **Diaphragm Valve:**

   Check for leakage from bonnet at spindle
   (Diaphragm leaking)
   Check flange for leakage

5. **Needle Valve:**

    Check gland for leakage

    Oil spindle at gland (except for gas, oxygen - acetylene application)

    Check line connections for leakage

6. **Check Valve:** (Non-return valve)

    Check dashpot for leaks

    Check trunnion seals

    Check trunnion bearings

    Lubricate trunnion bearing

    Check flanges for leakage

In addition to the foregoing preventive maintenance recommendations, reference must be made to the Manufacturer's instruction manual and name-plate data. Safety must be the overriding factor in all maintenance activities and this rule must be applied to the material in this pamphlet.

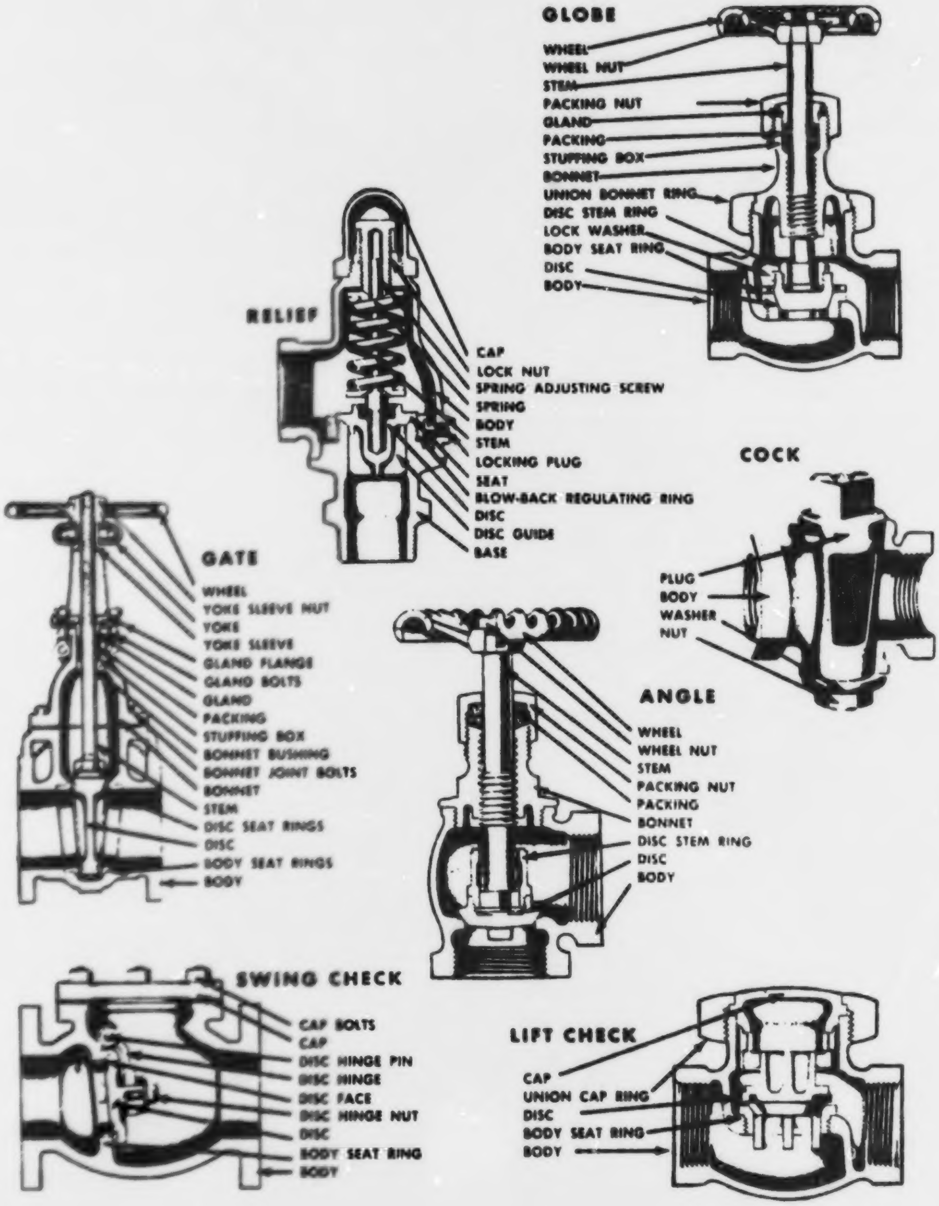


Figure 3-22 Basic Valve Designs

SAFETY

In the foregoing pages little reference was made to safe working procedures for work on pipelines. Here are a few "do's" and "don'ts" on safety when working on pipeline maintenance:

1. Before opening a pipeline be absolutely certain that you know what kind of fluid the line contained.
2. Make sure that the pipeline has been drained and, whenever necessary, purged and/or vented.
3. Lock in the closed position the connecting valves on either side of the section to be opened and attach a signed and dated danger tag to each valve.
4. When opening the initial flange do so by backing off the flange fasteners sufficient to separate the two flange halves just enough to confirm that the pipeline is not under pressure.

   Never remove all the fasteners and then separate the flange halves, to do so is to invite disaster.

5. Exercise extreme caution in your approach to pipeline maintenance:
   a. Don't be misled by so-called low pressure systems, they too can pack a mighty punch for the unwary.
   b. Don't rely on others for your protection, you cannot depend on anyone but yourself.

    c. Don't take chances, if you are not sure, seek the advice of your superiors.
    d. Do wear your protective clothing and equipment. Intelligent people make use of theirs.

6. When the job is completed, notify the operator in charge. Remove padlocks and danger tags from the valves. Return tools and equipment to the respective storage areas, tag faulty tools and equipment for repair. Leave the job site in a clean, tidy and safe condition, not necessarily as you found it, but as you would have liked to have found it.

Isolated Sludge Pockets

The duplication of sludge handling equipment and pipelines to provide increase flexibility and to ensure continuity of operations may have created an unexpected and hazardous situation. Whenever sludge is isolated in sections of pipeline between two closed valves, bacterial action produces methane and a corresponding increase in pressure. Unless provision is made for the relief of the pressure, it will increase to a level at which the pipe, or fitting, will yield. The pipe will burst with violence and should anyone be as unfortunate as to be in the vicinity, they may be injured or worse.

Illustration 1 shows a section of pipeline in which sludge can be trapped between two closed valves. Illustration 2 shows a section of pipe, at a pump, in which sludge might be trapped between the neck valve and the gate valve. Explosions have occurred in two identical locations at two separate sewage treatment plants. Fortunately, no one was present at the time of the explosions.

The only sure way to avoid explosions from this cause is to drain and flush the pipeline. Failing that, pipelines containing sludge should always be open at one end, or a vent provided as in Illustrations 3 and 4. Any gas vented off the lines would be carried to the outside of the building and clear off any possible source of ignition.

Illustration 1

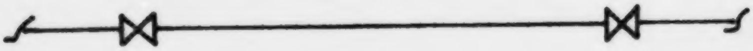


Illustration 2

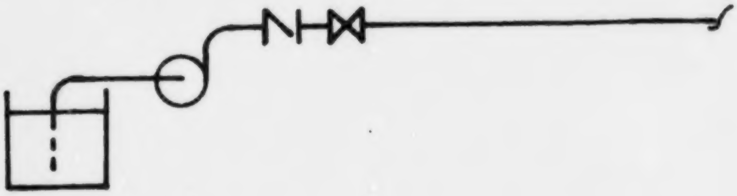


Illustration 3

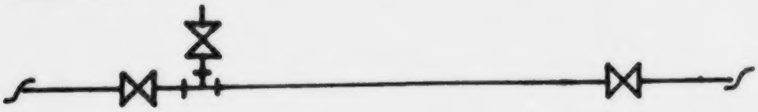
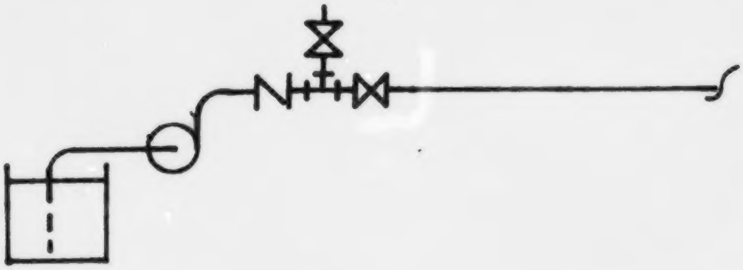


Illustration 4



One final comment, always obey rules and regulations. They were not introduced by well meaning people with little else to occupy them. For the most part, safety rules and regulations came about as a result of the investigation of accidents in which people have been injured or even killed.

SUBJECT: **TOPIC: 4**

PREVENTIVE MAINTENANCE **LUBRICATION**

OBJECTIVES:

The trainee will be able to:

1. **Recall the functions of a lubricant.**
2. **Recall the important aspects of storage and handling of lubricants.**
3. **Recall the explanation of the following characteristics of oil:**
   - (a) viscosity
   - (b) pour point
   - (c) oxidation resistance
   - (d) water separation
   - (e) oil type
   - (f) flash point
4. **Recall the explanation of the following characteristics of grease:**
   - (a) hardness
   - (b) pumpability
   - (c) dropping point
   - (d) water resistance
   - (e) stability

LUBRICATION

PURPOSE OF LUBRICATION

The main functions of a lubricant are to reduce friction and wear.

Friction is resistance to motion which arises when a surface slides on another surface.

Wear is any loss or destruction of the surface as a result of the sliding.

To understand how a lubricant reduces wear, imagine two blocks with smoothly ground surfaces, one on top of the other. If we pull on the top block with a certain force, we find it requires a pull of perhaps 15 lbs. to move the top block. However, by placing some oil between the blocks, we find that a pull of only 2 1/2 lbs. is required to slide the top block. The oil has reduced the friction between the two blocks.

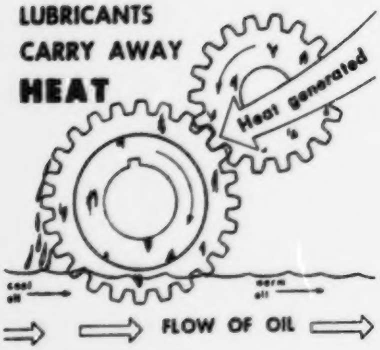
What causes friction? If one were to look at the surfaces of the steel under a powerful microscope, one could detect a definite roughness or minute peaks and valleys. This is what prevents easy sliding. However, when oil is present between the blocks, the blocks are separated and the small peaks and valleys can no longer interlock. The top block slides very easily over the bottom block.

What causes wear? Many clean metal surfaces tend to stick together if brought into direct contact. Usually air, grease or oxide interfere with establishing this contact. However, as surfaces rub together, the masking films may be torn away momentarily. The areas of metal thus freshly exposed to direct contact will tend to cling together or even weld fast. This sets up a resistance to sliding motion. Since wear is usually associated with friction, the high areas of the two surfaces rub each other. This wearing process tears away fragments and the surface becomes rougher and the rate of wear increases and the loosened fragments may hasten wear by scratching and gouging the surfaces.

Friction and wear cannot be eliminated but can be reduced by the use of the correct lubricant.

Other Important Functions of Lubricants

1. Carry away heat (Oil)



Another function of liquid lubricants is that of carrying away HEAT. Heat in a machine is developed at the points of contact between the rubbing and rolling part by the medium handled (hot water) and as a result of rolling friction within the lubricant. In some circulating systems, the hot oil is carried off to coolers and then returned for another cycle. In some systems, the heat eventually passes from the oil, through the sides of the oil reservoir, and into the air. Therefore, it is normal for the oil in reservoirs to become heated when the machinery operates, but **EXCESSIVE** heating of the oil is a danger signal that calls for an inspection of the machine and the lubricant to determine the cause of the excess heat.

2. To aid in sealing against dirt.

The proper use of a lubricant will keep the bearings clean. In this drawing, the sleeve bearing has been filled with grease and the excess has come out at the ends to form a seal around the shaft. Each time a new addition of grease is applied at the middle of the bearing, it replaces the lubricant, cleanses the bearings and maintains the seal.

3. To protect metal surfaces against rust and corrosion.

STORAGE AND HANDLING

Proper storage and handling of lubricants is essential to good maintenance. Every plant, regardless of its size, should have a definite place for storage of oils and greases, even if it is only in a designated corner. Larger facilities should have a centrally located oil storage house or a special room. This room or building should be of fireproof construction, with brick or concrete walls, steel beams and a metal or concrete roof.

Exposed heating devices should be placed so that it is impossible to store lubricants against them. Wide fluctuation of temperatures should be avoided since it can lead to condensation of moisture within the lubricant containers. Approved fire extinguishers should be readily available.

Cleanliness and orderliness are two prerequisites for oil storage. The storage area should be organized, i.e. racks for oil drums, space for grease drums, shelves for smaller containers and an especially clean area or cupboard for oil cans, grease guns, etc. A closed metal container must be provided for discarded oily rags, etc.

Lubricant containers should be kept clean on the outside and lubricant pumps, measuring cans, grease guns, etc., should be stored away from open windows to avoid contamination from dirt, dust or other foreign objects. When oil drums are stored on end, it is important to see that water does not get inside through loose or open bung holes. If paddles are used to scoop grease from the container they should be of metal fabrication.

CHARACTERISTICS OF LUBRICANTS

Proper lubrication has been defined as the use of the right lubricant in the right amount at the right place at the right time. In selecting the proper lubricant, the design of a machine and its operating conditions must be considered. Among the facts necessary to consider are:

1. Recommendations of the lubricant manufacturer.
2. Method of lubrication.
3. Operating speed and load.
4. Type of bearings.
5. Operating and ambient temperature.
6. Frequency of lubrication.

The following paragraphs discuss the characteristics of oil and grease which are considered when selecting a particular lubricant.

OIL

The characteristics of an oil of importance in lubricant selection are:

1. Viscosity
2. Pour Point
3. Oxidation Resistance
4. Water Separation
5. Oil Type
6. Flash Point

Viscosity

Viscosity is the measure of flowability or thickness of a liquid, at definite temperatures. Low-viscosity oils, often referred to as light oils, flow freely. High-viscosity or heavy oils, flow slowly. Oils having viscosities between these extremes are called medium-viscosity oils. See Figure 4-1.

In general, heavy oils are used on parts moving at slow speeds under high pressure, since the heavy oil better resists being squeezed out from between the rubbing parts. Light oils are used when higher speeds and lower pressures are encountered, as they do not impose as much drag on high speed parts and the high speed permits a good oil wedge to form even though the oil is light. Low temperatures call for light oils and high temperatures are more apt to demand the heavier oils because oils increase in viscosity as their temperatures become lower and decrease in viscosity as the temperatures go up. An exception to this is multi-grade oil.

Pour Point (Figure 4-2)

Pour point is another property of oil that is particularly important if the oil is to lubricate machinery operated in unheated buildings or outdoors during cold weather. The pour point of oil is the lowest temperature at which it will pour, or flow when chilled without disturbance. Even as water loses its fluidity and becomes solid at a certain temperature, so too, does oil cease to flow. Some oils with high pour points will stop flowing at 40°F whereas others with low pour points will still flow freely in temperatures below zero.

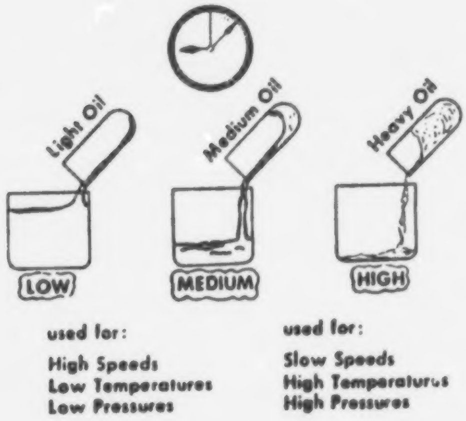


Figure 4-1 Viscosity

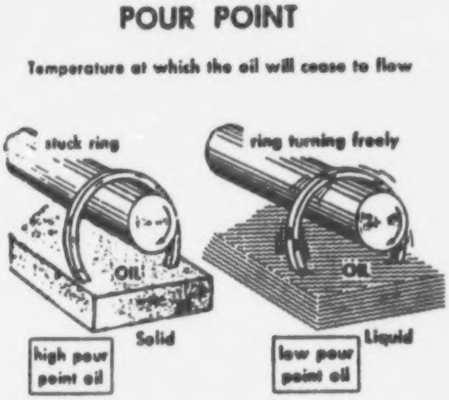


Figure 4-2 Pour Point

Oxidation Resistance

Oil is a complex mixture of compounds of hydrogen and carbon called hydrocarbons. When hydrocarbons are exposed to air and heat, they combine slowly with the oxygen in the air and are chemically changed into materials unsuitable for use as lubricants. The slow combination of oil with oxygen is called oxidation, and the materials formed are called the products of oxidation.

While oxidation occurs slowly when oils are in service, churning and splashing of the oil and exposure of the oil to temperatures much above room temperature hastens oxidation. Turbines and large circulating systems, in which oil is used for long periods without being changed, must have oils with high resistance to oxidation. These are more expensive because their refining costs are higher. Where oil remains in service only a short time, or new oil is frequently added to make-up, those grades with lower oxidation resistance may serve satisfactorily.

Water Separation (Figure 4-3)

When water enters and is churned up with oil, an emulsion is formed which has poor lubricating properties. If the oil is clean, it will separate from the water if it is allowed a rest period. This characteristic is known as demulsibility. Demulsibility of the oil is good if it separates quickly and poor if it separates slowly. It should be noted here that the presence of products of oxidation may slow down or stop this separating process.

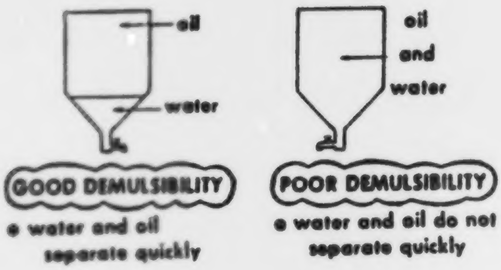

oil
water
GOOD DEMULSIBILITY
• water and oil separate quickly
oil and water
POOR DEMULSIBILITY
• water and oil do not separate quickly

Figure 4-3 Water Separation

Oil Type

Extreme-pressure oils are used on gears and bearings that operate under extreme loads and pressures. With extreme shocks and pressures, moving steel parts may break through an ordinary oil film and establish steel-to-steel contact. The EP (extreme pressure) oils contain added ingredients that help prevent metal-to-metal contact even if the oil film itself should fail to maintain the necessary separation.

Oils lubricating the insides of steam cylinders are exposed to both heat and water. These oils have an animal-fat oil to them to assist them in maintaining a lubricating film on the hot, wet walls of the steam cylinder. Such a fatty oil is called a **COMPOUNDED OIL.**

Oils used to lubricate internal combustion engines and air compressor cylinders have a detergent (cleaning agent) added to them which helps keep deposits from forming on the piston rings and other surfaces in contact with the oil. Such an oil is known as a DETERGENT OIL.

Excessively loaded gears need
Extreme Pressure Oil
(E.P. Oil)

Steam Cylinders need
Compounded Oil

Diesel Engine
crankcase lubrication calls for
Detergent Oil

Flash Point

Flash point is the lowest temperature at which there are sufficient "vapours" to cause an explosion.

GREASE CHARACTERISTICS

The characteristics of grease which will be considered are:

1. Hardness
2. Pumpability
3. Dropping Point
4. Water Resistance
5. Stability

Hardness

Hardness, sometimes known as consistency or penetration, is a measurable characteristic of grease that must be considered in selecting a suitable grease for a particular lubrication job. No. 0 grease is soft, the hardness increasing to No. 1 and No. 2 and so on up to No. 6, which has the consistency of a cake of soap. Just as the soil in your garden is hard if a spade barely enters it and soft if it spades easily, so grease hardness is determined and graded according to the depth to which a sharp, pointed cone will penetrate the grease when dropped from a certain height.

Pumpability

Another important characteristic of a grease is the ease with which it can be pumped. For example, in centralized grease systems, the pump must develop sufficient pressure to force the grease through the lines and through the small passages in the numerous distribution valves to the machine bearings.

Two different greases that may be pumped with equal ease at ordinary room temperature will often differ widely in pumpability at a much lower temperature. The pressure gauge on the centralized system may show that the pump builds up a pressure of 1,000 psi when pumping grease (a), whereas a pressure of 2,000 psi is required to pump grease (b). The difference is due to the kinds and proportions of oils and soaps in the composition of the two greases.

A centralized grease system is designed for operation within a definite pressure range that should not be exceeded. Therefore, grease pumpability is of very real importance.

Dropping Point

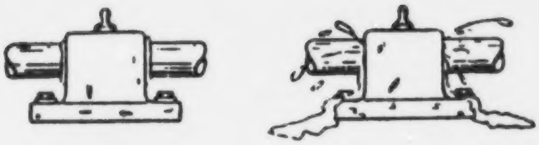
Different greases melt at different temperatures because of the soaps they contain. The Dropping Point of a grease is the temperature at which it passes from a semi-solid to a liquid. Most calcium (lime) soap greases melt in the 160-210°F range. Sodium soap greases melt in the 275-350°F range, whereas lithium soap greases melt in the 350-400°F range. For this reason, sodium and lithium soap greases are usually used where it is known that temperatures will be high.

Water Resistance

Some greases have to do a lubrication job in spite of contact with water, so that water resistance is an important property of greases. Greases with calcium and lithium soaps will not dissolve in water, but soda-soap greases are not used where there is a possibility that water will contact the lubricant. Fine, powdered graphite is added to some greases to improve their performance on plain bearings in the presence of water.

Stability

Some greases, when placed in a bearing, will retain their original hardness; that is, they are stable. Such greases will stay in antifriction bearings very well and give good lubrication for a long time. Other greases lose their hardness after being worked in a bearing for some time and become quite soft. They may even run out unless the bearing is very well sealed. Obviously, the type of grease that retains its hardness and stays in the bearing is the more desirable. Such a grease is said to have good stability. See Figure 4-4.



GOOD
Grease Stays in the Bearing

POOR
Grease Thins and Runs Out of the Bearing

Figure 4-4 Stability

Lubrication Chart

Because of the many differences in the properties of lubricants and in the machines to be lubricated, considerable knowledge and study are necessary to select the proper lubricant for a particular machine. The eventual choice is based on the specifications of the equipment manufacturer, the recommendations of the lubricant manufacturer and the particular conditions under which the machine will operate.

Lubrication charts or records should be prepared and kept up-to-date for all the different items of equipment. The chart should list the different parts of a machine to be lubricated, the method of application, and the type of lubricant to be used. If applicable, the frequency of lubrication should be recorded. The worst problem in industry today is not due to lack of grease, but too much grease. The theory is if one shot is good, five shots are better. This is totally incorrect. Test the bearing with your hand. If cool leave alone, it is okay. If not and it has been greased recently, remove plug or grease fitting to release excess grease.

LUBRICATION TERMINOLOGY

SAE -- Society of Automotive Engineers

SAE Viscosity Grades

Summer Grades

20 30 40 50 -- measured at 99°C (210°F)

Winter Grades

5W 10W 15W 20W - measured at -18°C (0°F)

AGMA -- American Gear Manufacturers Association

ISO -- International Standards Organization

API -- American Petroleum Institute

API Service Classification for Engine Oil Use

Gasoline Engines

SA	light duty	- straight mineral oil
SB	light duty	- minimum duty (non detergent)
SC	light duty	- meets 1964-67 requirements
SD	light duty	- meets 1968-71 requirements
SE	light duty	- meets requirements beginning in 1972
SF	severe duty	- meets requirements beginning in 1980

Diesel Engines

CA	light duty
CB	moderate duty
CC	service typical of highly supercharged diesels.
CD	high speed, high output supercharged diesels.

Old API Classifications

Gasoline		Diesel
ML	light	DG
MM	moderate	DM
MS	severe	DS

API Service Classifications for Gear Oil

API	GL - 1	light to moderate conditions
API	GL - 2	light to moderate conditions
API	GL - 3	light to moderate conditions
API	GL - 4	light to moderate conditions
API	GL - 5	extreme pressure use
API	GL - 6	extreme pressure use

Military Specification	API Service
MIL - L - 46152	SE
MIL - L - 2104A	CA, CB
MIL - L - 2104B	CC
MIL - L - 2104C	CD

SUBJECT: **TOPIC: 5**

PREVENTIVE MAINTENANCE **BEARINGS**

OBJECTIVES:

The trainee will be able to:

1. Recall the various types of bearings and their functions.
2. Recall the importance of cleanliness.
3. Recall procedures of mounting and dismounting.
4. Recall the considerations of variable temperature conditions for the different types of bearing applications.
5. Recall the lubrication factors for bearings.
6. Recall the common causes of bearing failures.

BEARINGS

TYPES AND FUNCTIONS

General

Designers have a wide variety of bearing types from which to choose. Each bearing type has characteristics which make it the best choice for a certain application. The original selection is made to assure maximum bearing performance. Therefore, a replacement should conform to the manufacturers specifications. The following resumé of the functions of each bearing type will aid in selecting replacements. See Figure 5-1.

Self-Aligning Ball Bearings

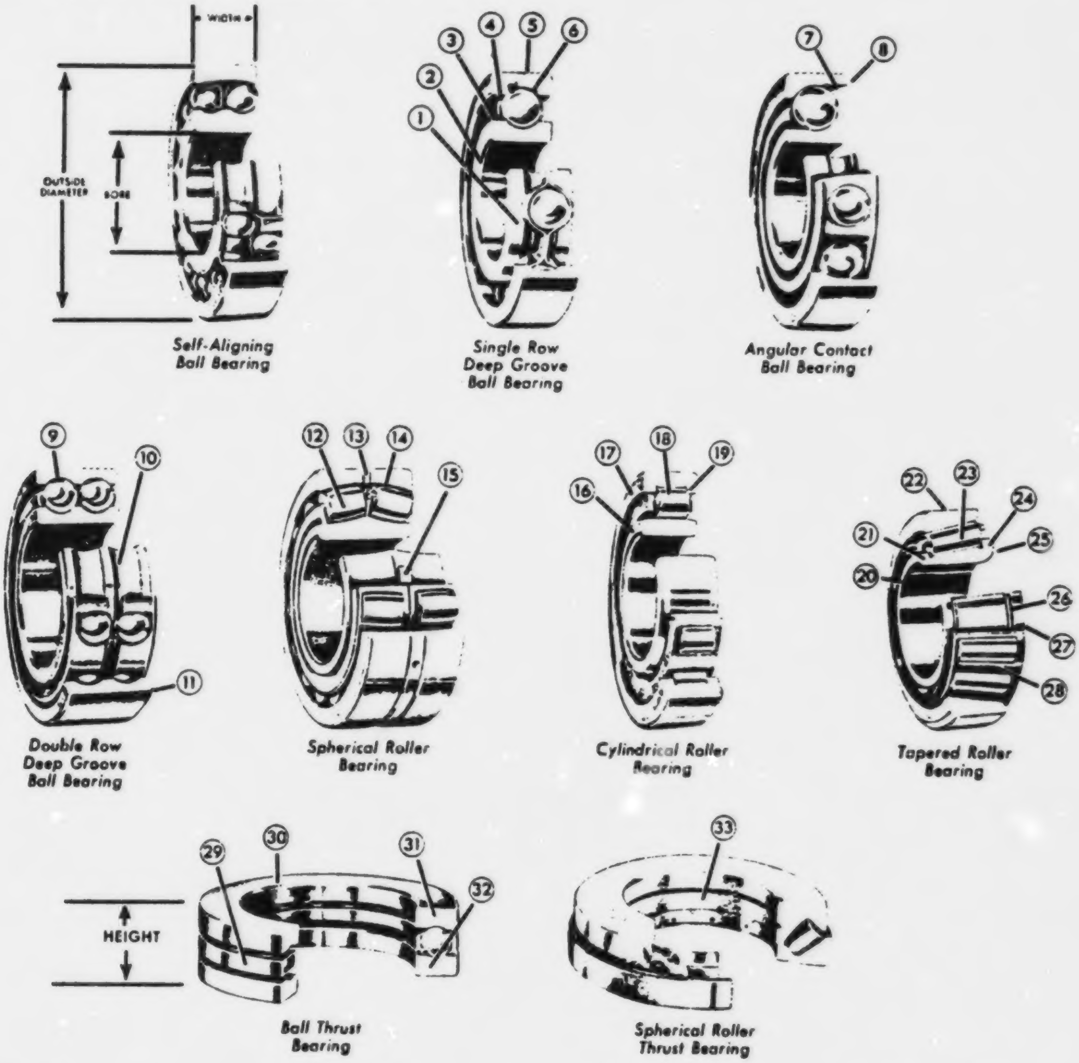
The self-aligning ball bearings, with two rows of balls rolling on the spherical surface of the outer ring, compensates for angular misalignment resulting from errors in mounting, shaft deflection and distortion of the foundation. It is impossible for this bearing to exert any bending influence on the shaft, a most important consideration in applications requiring extreme accuracy, at high speeds. Self-aligning ball bearings are recommended for radial loads and moderate thrust loads in either direction.

Single Row, Deep Groove Ball Bearings

The single row, deep groove ball bearing will sustain, in addition to radial load, a substantial thrust load in either direction, even at very high speeds. This advantage results from the intimate contact existing between the balls and the deep, continuous groove in each ring. When using this type of bearing, careful alignment between the shaft and housing is essential. This bearing is also available with deals and shields, which serve to exclude dirt and retain lubricant.

The illustrations below identify the bearing parts of the nine SKF basic bearing types. The terms used conform with the terminology section of the AFBMA* Standards—and are mutually accepted by the anti-friction bearing manufacturers.

***Anti-Friction Bearing Manufacturers Association, Inc.**



Self-Aligning Ball Bearing

Single Row Deep Groove Ball Bearing

Angular Contact Ball Bearing

Double Row Deep Groove Ball Bearing

Spherical Roller Bearing

Cylindrical Roller Bearing

Tapered Roller Bearing

Ball Thrust Bearing

Spherical Roller Thrust Bearing

1. Inner Ring
2. Inner Ring Corner
3. Inner Ring Land
4. Outer Ring Land
5. Outer Ring
6. Ball
7. Counter Bore
8. Thrust Face (Face)
9. Outer Ring Raceway
10. Inner Ring Raceway
11. Outer Ring Corner
12. Spherical Roller
13. Lubrication Feature
14. Spherical Outer Ring Raceway
15. Guide Ring
16. Inner Ring Side
17. Outer Ring Side
18. Cylindrical Roller
19. Locating Rib
20. Cone Front Face
21. Cone Front Face Rib
22. Cup (Outer Ring)
23. Tapered Roller
24. Cone Back Face Rib
25. Cone Back Face
26. Under Cut
27. Cone (Inner Ring)
28. Cage
29. Ball Cage
30. Face
31. Small Bore Washer
32. Large Bore Washer
33. Sleeve

Figure 5-1 Bearing Nomenclature

Angular Contact Ball Bearings

The angular contact ball bearings supports a heavy thrust load in one direction, sometimes combined with a moderate radial load. A steep contact angle, assuring the highest thrust capacity and axial rigidity, is obtained by a high thrust supporting shoulder on the inner ring and a similar high shoulder on the opposite side of the outer ring. These bearings can be mounted singly; or, when the sides are flush ground, in tandem for constant thrust in one direction; mounted in pairs, also when sides are flush ground, for a combined load, either face-to-face or back-to-back.

Double Row, Deep Groove Ball Bearings

The double row, deep groove ball bearings embodies the same principle of design as the single row bearing. However, the grooves for the two rows of balls, may be so positioned that the load lines through the balls have either an outwardly converging or an inwardly converging contact angle. This bearing has a lower axial displacement than occurs in the single raw design.... substantial thrust capacity in either direction....and high radial capacity due to the two rows of balls.

Spherical Roller Bearings

The spherical roller bearing, due to the number, size and shape of the rollers, and the accuracy with which they are guided, has unexcelled capacity. Since the bearing is inherently self-aligning, angular misalignment between the shaft and housing has no detrimental effect and the full capacity is always available for useful work. The design and proportion are such that, in addition to radial load, heavy thrust load may be carried in either direction.

Cylindrical Roller Bearings

The cylindrical roller bearing has high radial capacity and provides accurate guiding of the rollers, resulting in a close approach to true rolling. Consequent low friction permits operation at high speed. Those types which have flanges on one ring only, allow a limited free axial movement of the shaft in relation to the housing. They are easy to dismount even when both rings are mounted with a tight fit. The double row type assures maximum radial rigidity and is particularly suitable for machine tool spindles.

Ball Thrust Bearings

The ball thrust bearing is designed for thrust load in one direction only. The load line through the balls is parallel to the axis of the shaft, resulting in high thrust capacity and minimum axial deflection. Flat seats are preferred.... particularly where the load is heavy....or when close axial positioning of the shaft is essential; as for example, in machine tool spindles.

Spherical Roller Thrust Bearings

The spherical roller thrust bearings is designed to carry heavy thrust loads, or combined loads which are predominantly thrust. This bearing has a single row of rollers which roll on a spherical outer race with full self-alignment. The cage, centered by an inner ring sleeve, is constructed so that lubricant is pumped directly against the inner ring's unusually high guide flange. This ensures good lubrication between roller ends and the guide flange. The spherical roller thrust bearing operates best with relatively heavy oil lubrication.

Tapered Roller Bearings

Since the axes of its rollers and raceways form an angle with the shaft axis, the tapered roller bearing is especially suitable for carrying radial and axial loads acting simultaneously. A bearing of this type usually must be adjusted toward another bearing capable of carrying thrust loads in the opposite direction. Tapered roller bearings are separable their cones (inner rings) with rollers and their cups (outer rings) are mounted separately.

MOUNTING OF BEARINGS - SHAFTS AND HOUSINGS

Mounting Deigns

The successful application of a ball or roller bearing depends to a great degree on two factors:

1. the mounting design
2. the accuracy of the various machines surfaces upon which the bearing is supported

Unless care is exercised in both designing the bearing mounting and machining the various parts, a premature failure may result. This is true, even though the capacity of the bearing specified may be well above the applied load.

The narrow width of a ball or roller bearing provides a comparatively small surface area for the applied load. Hence, the unit pressures are high and must be uniformly distributed to the supporting surfaces of both the shaft and the housing. Therefore, it is evident that both the inner and outer rings must be properly mounted on the shaft and in the housing respectively. Serious distortion is eliminated and uniform distribution of the load is obtained.

This same care must also be used with respect to the pedestal on which the bearing housing is mounted. Distortion of the bearing housing will occur, unless the pedestal is firm and the surface straight.

A bearing housing should be designed to permit a simplified assembly and disassembly of the component parts. Such design should also provide for simple methods of casting, with few machined surfaces to reduce costs to a minimum.

Cleanliness

The importance of protecting the bearing from the intrusion of dirt and other foreign matter cannot be over emphasized. Seals must be designed to prevent the entrance of any type of foreign matter to the bearing. Dirt will act as a perfect lapping compound and will grind the bearing surfaces. This will result in excessive wear of the rolling elements and of the inner and outer rings. In addition, foreign matter will contaminate the lubricant and gradually break down to oil or grease to the point where effective lubrication cannot be obtained. Dirt will also cause noisy operation.

The cavity in the housing around the bearing should be large enough to provide for an adequate amount of lubricant. Seals should be effective in retaining the lubricant in the housing.

Mounting and Dismounting

Bearings may be mounted directly on the shaft or on tapered adapter sleeves.

When mounted directly on the shaft, the inner ring should be located against a shaft shoulder of proper height (refer to bearing manufacturer's specifications). This shoulder must be machined square with the bearing seat and a shaft fillet should be used. The radius of the fillet must clear the corner radius of the inner ring. This also applies when mounting the outer ring in the housing. Generally speaking, bearing applications have a rotating inner ring with a stationary load applied to the bearing. In such cases, it is necessary to mount the inner ring with a press fit on the shaft and the outer ring with a slip fit in the housing.

Creeping will occur when the inner ring is loose on the shaft. This will result in overheating, excessive wear and contact erosion between the shaft and the inner ring. The explanation of creeping action can be simplified by the use of this comparison. Compare the action of an internal gear in mesh with a planet pinion. If the planet pinion (representing the shaft), had one tooth less than the internal gear (representing the bearing inner ring), the latter would slip back one gear tooth in each full revolution of the shaft. Hence, the speed of the shaft (planet gear) would be greater than the speed of the inner ring (ring gear), and sliding would take place.

Therefore, it is an obvious conclusion that a preventive measure must be taken to eliminate creeping and its harmful results. Mount the inner ring with a sufficient press fit on the shaft! This will ensure that both inner ring and shaft act as a unit, and both rotate at the same speed. It is also desirable to use a locknut to clamp the inner ring against the shaft shoulder.

If the applied load is of a rotating nature, for example, vibrating screens where unbalanced weights are attached to the shaft, then the outer ring becomes the critical member. In order to eliminate creeping in this case, the outer ring must be mounted with a press fit in the housing. The rotating inner ring, when subjected to a rotating load, can be mounted with a slip fit on the shaft.

It is evident, therefore, that when the applied load is stationary with respect to the outer ring, a tight shaft fit is required. When the applied load is stationary with respect to the inner ring, a tight housing fit is required.

Some bearings may be pressed on the shaft by means of a brass tube. Pressure is applied to the inner ring of the bearing by the tubing. The assembly of larger bearings can be simplified by a "heating" procedure. Heat the bearing in clean oil to a temperature of approximately 90°C provided it is not prepacked and sealed. Care should be exercised not to exceed this temperature. Sufficient time should be allowed to assure that the bearing is heated throughout. This will sufficiently expand the bearing and permit it to slide easily over the shaft to its proper location. Care should be taken to prevent cocking of the inner ring as this may cause it to seize before the bearing is properly located. If this happens, it is best to remove the bearing promptly and reheat. When the bearing cools, it will shrink and grip the shaft. It requires only a little ingenuity to provide the tools to facilitate this procedure.

Shaft and Housing Fits

A ball or roller bearing has extremely accurate component parts which fit together with very close tolerance. The inner ring bore and the outer ring outside diameter must be manufactured within close limits to fit their respective supporting members - the shaft and the housing. It follows that the shaft and housing must also be machined to close limits. Only then, will we obtain the required fit when the bearing is mounted.

It is advisable to refer to the tables of shaft and housing fits as published in the bearing manufacturer's catalogue. These tables list recommended dimensions and tolerances for various operating conditions.

Adapter Sleeve

The use of a tapered adapter sleeve on standard shafting will greatly simplify the mounting of the bearing. The outer surface of the sleeve is tapered to match the tapered bore of the bearing inner ring. This will provide the required tight fit between the inner ring and the shaft. The adapter sleeve is slotted to permit easy contraction and is threaded at the small end to fit a locknut. When the sleeve is drawn-up tight between the bearing and the shaft, a press fit is provided both at the shaft and the inner ring.

A lockwasher between the inner ring and the locknut prevents the nut from loosening. The use of micronized graphite will greatly facilitate assembly, and help prevent contact erosion of the parts. A light coating of graphite is applied on the shaft, adapter sleeve and the bore of the inner ring. It is good practice to apply a coat of oil to the shaft, housing and component parts before the bearing unit is assembled.

Internal Bearing Fits

It is evident that a press fit between the parts will stretch the inner ring. This holds true when mounting the bearing directly on the shaft or by means of an adapter sleeve. Thus, there will be a tendency to reduce the initial internal radial clearance in the bearing.

If an excessive press fit is used, it is possible to reduce the bearing clearance sufficiently to make the bearing tight.

However, if the recommended shaft fits are used and normal operating temperatures prevail, the internal clearance remaining after mounting the bearing will be sufficient to ensure proper operation.

When an adapter mounting is used, some additional cautions should be heeded. If these are taken into consideration, the bearing will not be tight internally.

The following is recommended practice for mounting ball bearings on adapters. Slowly rotate and swivel the outer race by hand while tightening the locknut. A light drag in the bearing will be apparent as the inner ring stretches and the clearance is reduced. This is the point where further tightening of the nut is not desired. The nut should then be drawn up so that a prong of the lockwasher is opposite a slot in the nut.

Before mounting spherical roller bearings on adapters, the internal clearance should be checked and recorded. Insert a feeler gauge between the top roller and the outer ring. The bearing should be resting upright on a table when making this check.

After mounting the bearing on the shaft and adapter sleeve, the locknut is tightened sufficiently to reduce the initial clearance by the required amount. This is checked in the same manner as described, by using the feeler gauge. The decrease in internal clearance after mounting is found in the manufacturer's specifications as well as in the pillow block mounting instruction enclosure.

Temperature Conditions

Certain types of bearing applications require the consider ation of variable temperature conditions.

For example: An induced draft fan handling gases at high temperature. The fan wheel is mounted on the shaft. Therefore, that part of the shaft in the fan casing is subjected to the high temperatures. The bearings are usually mounted directly outside the fan casing. Thus, the temperature of the shaft at the bearing location is considerably higher than the room temperature. It follows that the inner ring of the bearing (mounted on the shaft) will be considerably hotter than the outer ring. This is true, as the outer ring is mounted in the housing which is isolated from the fan and not subjected to the high temperature. As a result, the inner ring will expand more than the outer ring. This action reduces the internal clearance in the bearing, in addition to the reduction due to the tight fit on the shaft.

To compensate for this additional reduction of internal clearance in the bearing, some provision must be made. Bearings with an internal clearance greater than standard should be used. This additional clearance will compensate for internal clearance reduction due to the temperature differential between the inner and outer rings. There are other cases where bearings with internal clearance greater than standard are required. Among these are dryer cylinder bearings in paper machinery where steam passes through the hollow shaft shaker screens where tight shaft and housing fits are required -- high speed applications where the temperature rise in the bearing, due to speed and load, is abnormal.

If this additional internal clearance is not provided, the bearing will be excessively tight internally and cannot operate successfully. Excessive high operating temperatures and subsequent breakdown of the lubricant due to carbonization will occur.

Free and Held Bearings

A temperature variation will expand or contract the component parts of any machine. Due to this condition, it is essential that such parts be permitted to expand or contract without restriction. For that reason, only one bearing on any one shaft should be fixed axially in the housing (called a "held" bearing) to prevent axial or end motion.

All other bearings on that same shaft should have adequate axial clearance in the housing (referred to as "free" bearings). Such free bearings will permit the shaft to elongate and contract without restriction.

Generally, it is preferred to hold the bearing at the drive end. However, consideration should be given to the bearing load. In some cases, it is best to hold the bearing that carries the smaller radial load, especially where the "held" bearing is subjected to a thrust load of some magnitude. This will tend to equalize the distribution of the load on the bearings.

In gear reducers where herringbone gears are used, only one bearing on one of the shafts should be "held", usually the input shaft. It is evident that the "V" shape of the gear teeth will locate the other shaft axially.

When a flexible coupling is used to connect two shafts, a "held" bearing is required on each of the shafts, as a flexible coupling permits end motion of both shafts.

LUBRICATION

The bearing unit can usually be lubricated with a good grade of sodium base grease for normal loads, speeds and operating conditions. The bearing housing should be filled approximately one-third full with grease.

An excessive quantity of grease will "build up" internal pressure in the bearing housing and result in high temperatures, and subsequent damage to the bearing.

Here is a safety measure to combat this danger when adding lubricant to the bearing. After the lubricant is added, remove the plug (preferably at the lower half of the housing) while the bearing is in operation. This permits the excessive lubricant to flow out of the housing and relieve the internal pressure. When the flow of grease ceases, be sure to replace the plug.

Speed and other operating conditions, in some cases, require that oil lubrication be used. In that event, the bearing housing should be equipped with adequate oil fittings. Such fittings should provide a static oil level at the centre of the lowest ball or roller in the bearing.

Either pipe fittings, commercial oil cups or glass gauges can be used. However, it is preferable to use sight glass oil gauges which can be clearly marked for static and operating levels. The operating level is slightly lower than the static level due to agitation of the oil.

The static level can be marked on the gauge before the unit is assembled. The operating level can be determined only after the bearing is in operation. The gauge should also be marked for the operating level and the lubricant, at no time, should be permitted to drop below this point.

Do's and Don't's on Lubrication

Follow instructions when furnished with machinery. Use only grease where grease is specified and oil where oil is specified. Be sure to use exactly the kind of lubricant the instructions call for.

Store grease in clean containers. Handle grease with clean paddles or grease guns. Keep grease containers covered.

Don't overfill. Grease or oil will ooze out of overfilled housings past seals and closures, collect dirt and cause trouble. Too much lubricant will also cause overheating. This is particularly true of bearings running at high speeds where the churning of the lubricant will cause the bearings to run excessively hot.

Don't permit any machine to stand around for months without turning it over once in a while so that all surfaces of the bearings will be covered with lubricant. Oil tends to drain down off a standing bearing and moisture may condense in the housing and cause corrosion.

Inspect seals and vents regularly.

BEARING FAILURES AND THEIR CAUSE

Just as a doctor cannot effect a lasting cure unless he knows the cause of an illness, the maintenance person must know the reasons for a bearing failure before he can correct such a condition.

While ball and roller bearings have proved by long experience to be much more reliable and to need less attention than bearings of other types, the principles governing their use are necessarily quite different from those applicable to plain bearings. Therefore, it is of utmost importance to draw the right inference from bearing failures.

Sometimes, this can be done very quickly by a casual inspection of the bearing and the condition surrounding the application. Other times, it is necessary to make a complete laboratory examination before the cause can be determined. In addition, there are times when the condition of the filed bearing is such that the actual cause of failure cannot be definitely established and an exhaustive study of the application and all conditions surrounding it must be made.

Dirt and Other Foreign Matter

One of the common causes of bearing failure is dirt. Bearings which fail because of dirt can be recognized in several ways. Generally, fine dirt in a bearing acts as an abrasive and causes lapping of the bearing parts. This can be seen and felt in the ball or roller paths of the self-aligning type bearing. These paths will be polished as will the balls and rollers. If there are no breaks in the surfaces of the rolling elements and the ball or roller path is clearly defined, then you can be sure that fine dirt was in it during service.

Bearings of the deep-groove type are not as easily examined since the ball paths and the balls cannot be readily seen. A preliminary check can be made by feeling the amount of clearance between the inner and outer races. This can be done by laying the bearing on a flat surface and holding one ring stationary. If the other ring can be moved in a position parallel to the flat surface, wear has taken place. Do not attempt to feel wear in a bearing when holding it in your hands. In this position, it is possible to feel the normal internal clearance which might be mistaken for wear.

Ball and roller bearings do not wear, even after long service, if they are kept clean and are lubricated properly. When they have reached their fatigue limit, flaking of the loaded areas occurs. This flaking is not considered wear in the same sense that we consider wear caused by abrasives. Many worn bearings have to be discarded long before the fatigue limit of the steel is reached.

If water or other corrosive agents enter the bearing, wear will occur very quickly. The iron oxide formed by rust is an abrasive and will lap or grind the balls or rollers undersize. Sometimes a bearing will become corroded before it is put in service. A motor or pump or some other machine may be flooded or water may be introduced with the lubricant. In a case of this kind, corrosion pitting will occur and is distinguished by its dark brown colour. This pitting will be where the balls or rollers contact the races. It will cause the bearing to be noisy and since the machine has just been put into service, the user will immediately blame the bearing.

If examination of a bearing indicates that wear has taken place, the actual application should be carefully checked. You may find a quantity of hard gritty particles such as produced by a grinding wheel or a buffing stand. Seals should be checked thoroughly to make sure that dirt is not going past the seals and into the bearing. The lubricant should be checked as well as the method of lubricating the bearings. Slipshod maintenance methods could be the reason for dirt getting into the lubricant and then passed on into the bearing.

Sometimes a simple, inexpensive shroud or sheet metal box can be used to deflect the dirt away from the bearing housing. This will pay for itself many times over.

The same conditions are obtained as a result of water or other corrosive agents entering the bearing housing. Everything possible should be done to deflect such fluids away from the openings of the bearing housing.

Foreign matter may get into the bearing in the form of chips or particles of steel because of failure of another part of the machine, such as a gear. Such chips in the ball path will be pressed into the races and balls or rollers, causing dents or scratches.

Inadequate or Improper Lubrication

A common cause of early bearing failure is insufficient lubrication. The amount of lubricant required by a ball or roller bearing is small. That is why so many of these bearings, unlike the majority of plain bearings, can be lubricated with grease. While the amount required may be small, the quality should be the best obtainable.

Some applications involving high temperatures, high speeds, very heavy loads, etc., require oil lubrication. In some instances, special methods are used for oil lubrication, such as "circulating oil", "oil mist", "drop feed", etc.

Lack of lubricant is revealed by a high pitched sound and a rapid rise in temperature (a rise in temperature can also be the sign of over-lubrication). Failure to correct this fault immediately is apt to be followed by damage. Intense overheating will draw the temper of the bearing steel, the parts will become soft and the bearing will fail very quickly. This condition can quickly be recognized by the colour of the bearing which in the early stages will be blue. If the bearing is forced to continue to operate, it might seize and result in damage to other parts.

Lack of lubricant may cause wear of the cage accompanied by smearing of the balls or rollers and their tracks in the rings.

Grease whose consistency is too solid, or oil that is too thick, is capable of producing a braking effect powerful enough to make the rolling elements slide. Such sliding will cause smearing of the effect parts and early failure occurs.

Very rapidly starting of a machine is an adverse condition for a rolling bearing. Where the load is heavy, a condition such as the one described in the foregoing paragraph occurs. This smearing takes place because of the sliding of the parts before the running temperature has been reached and while the grease is still solid. This is particularly true where bearings operate under low temperature conditions. Where the load is light, the inner race slides or skids on the balls or rollers before they have a chance to rotate and smearing again takes place. Under both conditions, the bearing will continue to operate but will become noisy and eventually will fail.

Incorrect Machining of Shafts and Housing

If the specified shaft diameter is exceeded, or the housing bore undersize, the internal radial clearance of the bearing may be removed due to the expansion of the inner ring and contraction of the outer ring.

Both of these conditions will cause an overload of the bearing, and flaking (it is sometimes known as spalling) of the races or balls takes place. This condition may be recognized by the appearance of the parts under load, their surfaces are chipped and flaked. It differs from wear since it is rough, rather than smooth. This is the normal way for a bearing to fail after long service under load. If it takes place in a short period of time then the bearing has been overloaded by some outside cause.

Normal fatigue failures generally show a load track around the circumference of one ring and over one section of the other. Should the load track appear at two or more places in the one race, then internal loading is the cause.

For example, if the bore of the housing is out of round or undersize, the outer race will be squeezed out of shape, causing overloaded. In a cartridge mounting, the housing bore may be correct when disassembled but if the cartridge is put into a machine frame which is bored undersize, it may be squeezed out of round.

Distortion of the housing bore may occur if the pedestal on which the pillow block is bolted is not straight or smooth. The dimension of the housing may be correct, yet it is possible to pinch a bearing in a split housing if the bearing is not seated squarely in the bearing seat. If the cap is bolted down with the bearing in this position, pinching will take place across the diagonal corners of the outer ring.

The loading of bearings which are pinched in their housings because of cocking is very evident upon examination. A definitely defined load track will be noticeable on opposite sides of the outer race across the same diagonal where the pinching occurred.

Bearings mounted on a tapered shaft or on a tapered adapter sleeve can be overloaded at the time of mounting. If the inner race is forced up on the taper too far, it will expand excessively and the normal internal radial looseness in the bearing will be removed. When mounting ball bearings of this type, care should be forced up on the taper. When it starts to bind to the smallest degree, the inner race has expanded as much as it should. In a spherical roller bearing, both the amount of initial internal radial clearance and the necessary expansion of the inner ring, when mounted on a tapered shaft or sleeve, are such that the decrease in internal clearance can be measured with a feeler gauge. Thus, you can be sure the reduction is in accordance with specifications given in catalogues.

Extreme care must always be exercised to ensure proper shaft and housing fits. Undersized shafts and oversized housing bores will also cause early bearing failures. An undersized shaft will permit the inner ring to rotate on the shaft, while an oversized housing bore will permit the outer ring to spin in the housing. Both conditions will cause heat, smearing of the parts that rub, and rubbing cracks, all of which may lead to early failure.

Incorrect Mounting and Abuse

Misalignment of either race of a deep groove bearing will impose an undetermined load on the bearing. This load, together with the normal load on the bearing, will cause an overload resulting in early failure.

Misalignment also imposes undue stress on the retainer. In fact, this condition and poor lubrication are the main causes for retainer failures. If misalignment is present, it will be indicated by a ball track not parallel to the edge of the groove. When the misaligned path is in the outer race (in applications where the inner race rotates) then the bore of the housing is not parallel to the shaft. If the load path in the inner ring is not parallel with the edge of the groove, it is generally due to the inner ring being cocked on the shaft, the shaft shoulder not square with the bearing seat, or the shaft being bent.

Sometimes a condition like this can develop in the case of a cartridge housing mounting. The bearing may be mounted perfectly in the cartridge but if this unit is not bolted squarely to the machine frame, misalignment may result.

Another cause of initial overloading is an opposed mounting which binds the outer race and causes excessive thrust. Most mounting designs require that one bearing be held within its housing shoulders while all other bearings on that same shaft are allowed to be free to float in the housing. However, there are mounting designs which require that the outer races of two bearings be opposed, i.e. opposite sides of the two bearings are mounted against the shoulder of each housing. The spacing between these housing shoulders must be quite accurate in relation to the distance between the bearing shoulders on the shaft. Otherwise, excessive initial end thrust will be imposed on the bearing.

Pitting due to electric action causes noisy bearing operation. This occurs in electric motors and generators when electric current passes through the bearing from the frame to the shaft. An actual crater is burned in the steel by the arcing of the electric current. When the condition is very bad, these craters can be seen with the naked eye. Smaller ones require high magnification.

Bearings can become ball dented and when operated in this condition will be very noisy. Ball denting or brienelling is caused by excessive pressure on the races through the balls when the bearing is not rotating. It commonly occurs in small bearings that are mounted in the housing while the inner race is on the shaft. If the outer race is tight or becomes cocked in the housing and pressure to force it into position is applied through the inner race, such impressions are produced.

When the impact is in an axial direction, the balls can be forced against the edge of the groove which in turn, nicks them. Ball dents are easily distinguished from dents produced by chips or dirt because they are usually spaced in the same distance apart as the position of the balls. Ball dents can be produced in the bottom of a groove if sufficient impact goes through the balls in a radial direction. If a motor is dropped from a shelf to the floor and the overhanging end of the shaft strikes first, then denting of this nature can take place.

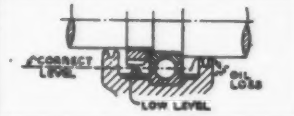
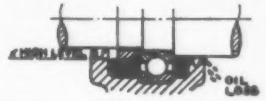
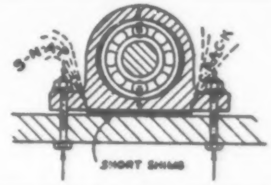
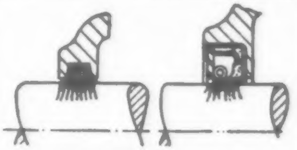
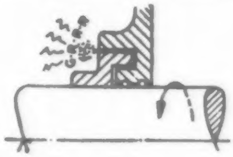
Another condition similar to ball denting is known as pocketing or false brinelling. In this condition, the pockets are also spaced the same distance apart as are the balls or rollers and because of this, it is often confused with denting. The cause, however, is entirely different and is due to vibration while the bearing is not rotating. If a shaft is supported by the bearings in transit and the parts are not bound tightly together, small pockets can be produced in the tracks. Such false brinelling is also likely to take place if two separate units are mounted on a common base and only one is operated at a time. The vibration caused by running one unit will cause the balls or rollers of the other, which is not running, to jiggle against the races and produce these small pockets. They can easily be distinguished from ball dents, especially if the race is examined with a magnifying glass, as the polishing lines in the race have been worn away at each pocket. On the other hand, ball dents depress the surface (deform it), and the polishing lines will continue on through these depressions.

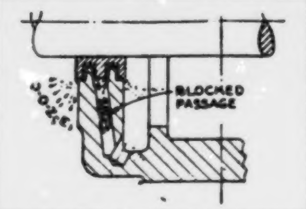
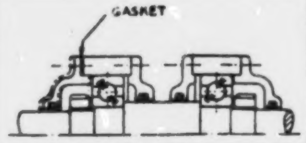
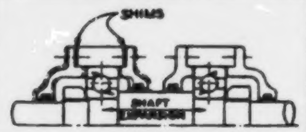
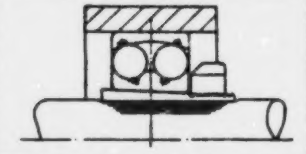
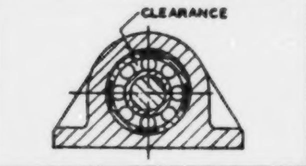
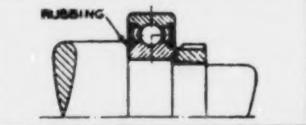
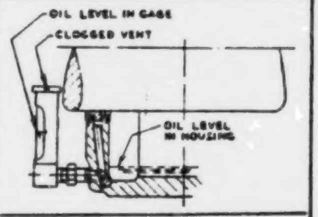
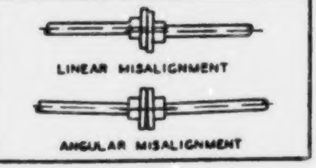
A common claim on the part of a customer is that a bearing has a flat ball. When such bearings are examined, it is found that the noisy condition which the customer thought was caused by a flat ball is due to something entirely different. It is extremely rare that defects in material or hardening are the cause of early bearing failure or noisy bearings. Another common complaint is a cracked inner race of small bearings. In most instances, it will be found that the shaft is oversized and the inner race was expanded excessively. It is possible that the inner race may have been forced on the shaft while it was in a cocked position.

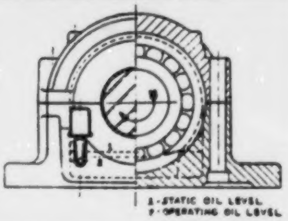
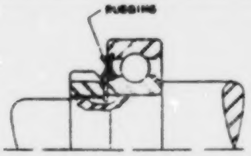
Bearings, though in common use every day, are precision units. They must be treated with common sense and care if they are to perform as expected.

TROUBLESHOOTING

OVERHEATED BEARING

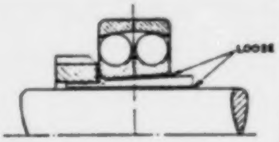
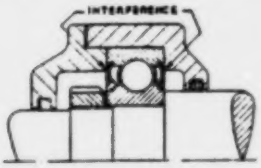
	REASON FOR CONDITION	PRACTICAL SOLUTION	
1	Wrong type of grease or oil causing break-down of lubricant.	Consult reliable lubricant manufacturer for proper type of lubricant. Check manufacturer's instructions to determine if oil or grease should be used.	
2	Low oil level. Loss of lubricant through seal. Insufficient grease in housing.	Oil level should be at center of lowest ball or roller in bearing. Using grease, lower half of pillow block should be filled.	
3	Housing packed with grease, or oil level too high . . . causing excessive churning of lubricant, high operating temperature, oil leakage.	Purge bearing until only lower half of housing is filled with grease. Using oil lubrication, reduce level to center of lowest ball.	
4	Bearings selected with inadequate internal clearance for conditions where external heat is conducted thru shaft, thereby expanding excessively the inner ring.	Replacement bearing should have identical marking as original bearing for proper internal clearance.	
5	Housing bore out of round. Housing warped. Excessive distortion of housing. Undersized housing bore.	Check and scrape housing bore to relieve pinching of bearing. Be sure pedestal surface is flat, and shims cover entire area of pillow block base.	
6	Felt too tightly packed in felt groove. Leather or composition seals with excessive spring tension or dried out.	Stretch felt to reduce friction. Replace leather or composition seals with ones having reduced spring tension. Lubricate seals.	
7	Rotating seals rubbing against stationary parts.	Check running clearance of rotating seal to eliminate rubbing. Correct alignment.	

	REASON FOR CONDITION	PRACTICAL SOLUTION	
8	Oil return holes blocked—pumping action of seals cause oil leakage.	Clean holes. Drain out used oil—refilling to proper oil level with fresh lubricant.	
9	Opposed mounting.	Insert gasket between housing and cover flange to relieve axial preloading of bearing.	
10	Two "held" bearings on one shaft. Excessive shaft expansion.	Back off covers in one of the housings, using shims to obtain adequate clearance of outer ring, to permit free axial bearing motion.	
11	Adapter tightened excessively.	Loosen locknut and sleeve assembly. Retighten sufficiently to clamp sleeve on shaft but be sure bearing turns freely.	
12	Unbalanced load. Housing bore too large.	Rebalance machine. Replace housing with one having proper bore.	
13	Rubbing of shaft shoulder against bearing seals.	Remachine shaft shoulder to clear seal.	
14	Incorrect oil level. Result: no lubricant in bearing.	Clean out clogged hole to vent oil gauge.	
15	Incorrect linear or angular alignment of two or more coupled shafts with two or more bearings.	Correct alignment by shimming pillow blocks. Be sure shafts are coupled in straight line—especially when three or more bearings operate on one shaft.	

	REASON FOR CONDITION	PRACTICAL SOLUTION	
16	Incorrect mounting of constant oil level cup. (Too high or too low.) Cup located opposite rotation of bearing permitting excessive flow of oil, resulting in too high oil level.	Oil level should be at center of lowest ball or roller when operating. Locate cup with rotation of bearing. Sketch illustrates correct position of constant level oil cup with respect to rotation.	
17	Prong rubbing against bearing.	Remove lockwasher — straighten prong or replace with new washer.	
18	Bearing seat diameter machined oversize, causing excessive expansion of shaft and bearing inner ring, thus reducing clearance in bearing.	Grind shaft to get proper fit between inner ring of bearing and shaft.	
19	"Pounding-out" of housing bore due to soft metal. Result: enlarged bore . . . causing spinning of outer ring in housing.	Rebore housing and press steel bushing in bore. Machine bore of bushing to correct size. If loads are not excessive, tighter fit in housing, without the use of the steel bushing, may correct the trouble.	

NOISY BEARING

1	Wrong type of grease or oil causing break-down of lubricant.	See Item 1 - Overheated Bearing
2	Low oil level. Loss of lubricant through seal. Insufficient grease in housing.	See Item 1 - Overheated Bearing
3	Bearings selected with inadequate internal clearance for conditions where external heat is conducted thru shaft, thereby expanding excessively the inner ring.	See Item 1 - Overheated Bearin,
4	Foreign matter (dirt, sand, carbon, etc.) entering bearing housing.	Clean out bearing housing. Replace worn-out seals or improve seal design to obtain adequate protection of bearing.
5	Corrosive agents (water, acids, paints, etc.) entering the bearing housing.	Addition of a shroud and (or) flinger to throw off foreign matter.
6	Housing bore out of round. Housing warped. Excessive distortion of housing. Undersized housing bore.	See Item 5 - Overheated Bearing

	REASON FOR CONDITION	PRACTICAL SOLUTION
7	Failure to remove chips, dirt, etc. from bearing housing before assembling bearing unit.	Carefully clean housing, and use fresh lubricant.
8	Rotating seals rubbing against stationary parts.	See Item 7 - Overheated Bearing
9	Opposed mounting.	See Item 9 - Overheated Bearing
10	Two "held" bearings on one shaft. Excessive shaft expansion.	See Item 10 - Overheated Bearing
11	Shaft diameter too small. Adapter not tightened sufficiently.	Metallize shaft and regrind to obtain proper fit. Retighten adapter to get firm grip on shaft. 
12	Adapter tightened excessively.	See Item 11 - Overheated Bearing
13	Flat on ball or roller due to skidding. (Result of fast starting.)	Carefully examine balls or rollers, looking for flat spots on the surface. Replace bearing.
14	Rubbing of shaft shoulder against bearing seals.	See Item 13 - Overheated Bearing
15	Distortion of bearing seals.	Remachine housing shoulder to clear seal. 
16	Prong rubbing against bearing.	See Item 17 - Overheated Bearing
17	Incorrect method of mounting. Hammer blows on bearing.	Replace with new bearing. Don't hammer any part of bearing when mounting.
18	Interference of other movable parts of machine.	Carefully check every moving part for interference. Reset parts to provide necessary clearance.
19	Distorted shaft and other parts of bearing assembly.	Only in extreme cases should a blow torch be used to facilitate removal of a failed bearing. Care should be exercised to avoid high heat concentration at any one point so distortion is eliminated.

	REASON FOR CONDITION	PRACTICAL SOLUTION
20	Bearing seat diameter machined oversize causing excessive expansion of shaft and bearing inner ring, thus reducing clearance in bearing.	Grind shaft to get proper fit between inner ring of bearing and shaft.
21	Unbalanced load. Housing bore too large.	Rebalance unit. Replace housing with one having proper bore.
22	"Pounding-out" of housing bore due to soft metal. Result: enlarged bore . . . causing spinning of outer ring in housing.	Rebore housing and press steel bushing in bore. Machine bore of bushing to correct size. If loads are not excessive, tighter fit in housing, without the use of the steel bushing, may correct the trouble.
23	Bearing exposed to vibration while machine is idle.	Carefully examine bearing for wear spots separated by distance equal to the spacing of the balls. Replace bearing.

REPLACEMENTS ARE TOO FREQUENT

1	Wrong type of grease or oil causing break-down of lubricant.	See Item 1 - Overheated Bearing
2	Low oil level. Loss of lubricant through seal. Insufficient grease in housing.	See Item 2 - Overheated Bearing
3	Bearings selected with inadequate internal clearance for conditions where external heat is conducted thru shaft, thereby expanding excessively the inner ring.	See Item 3 - Overheated Bearing
4	Foreign matter (dirt, sand, carbon, etc.) entering into bearing housing.	Clean out bearing housing. Replace worn-out seals or improve seal design to obtain adequate protection of bearing.
5	Corrosive agents (water, acids, paints, etc.) entering the bearing housing.	Addition of a shroud and (or) flinger to throw off the foreign matter.
6	Housing bore out of round. Housing warped. Excessive distortion of housing. Undersized housing bore.	See Item 5 - Overheated Bearing
7	Failure to remove chips, dirt, etc. from bearing housing before assembling bearing unit.	Carefully clean housing, and use fresh lubricant.

SUBJECT:

PREVENTIVE MAINTENANCE

TOPIC: 6

CHARACTERISTICS OF MATERIALS PUMPED

OBJECTIVES:

The trainee will be able to:

1. Describe the important characteristics of materials to be considered in pump selection.

2. Discuss the problems materials can create in:

   a) Water Pumping
   b) Sewage Pumping
   c) Sludge Pumping
   d) Chemical Pumping

3. Recall the information on the

   a) Pump
   b) Suction Side
   c) Discharge Side
   d) Drive

   that a supplier should be given to assist in pump selection.

4. Define cavitation and describe its possible effects.

CHARACTERISTICS OF MATERIALS PUMPED

INTRODUCTION

Pumps are subject to more misuse, abuse and faulty application than virtually any other type of equipment. In selecting a pump for a particular purpose or in replacing an installed pump which has failed either because of initial poor selection or because its task has changed, the operator usually disregards pump design and the type of service for which it is intended.

This Topic and Topic ____ will discuss the various factors involved in pump selection. These include:

1. Material characteristics.
2. The system in which the pump is to be installed.

Manufacturers recognize the importance of these factors, and break their pump lines into types of service. Consequently, to obtain the proper pump for a specific job the supplier must be provided as much information as possible on the installation. A rough sketch of the system is a most valuable aid. The information he requires includes:

1. **Material Data**
    - a) Type to be pumped
    - b) Viscosity of the material
    - c) Temperatures including maximum and minimum acceptable temperature
    - d) Specific Gravity
    - e) Weight per gallon
    - f) Corrosiveness and abrasiveness
    - g) Percentage solids and size of particles

2. Pump Data
    - a) Capacity required
    - b) Allowable variation
    - c) Duty cycle including hours worked per day, week or month; continuous or intermittent operation
    - d) Experience with previous pump (if applicable)

3. Suction Side Data

a) Is suction flooded?

b) Vertical, horizontal and total length of suction line

c) Piping material

d) Diameter of suction line

e) Number of valves, tees and elbows in suction line

f) List of any other equipment on suction side

g) What is the pressure or vacuum?

h) What is variation in suction pressure?

4. Discharge Side Data

a) Back pressure at pump.

b) Pressure variation.

c) Vertical, horizontal and total length of discharge line.

d) Piping material and diameter.

e) Number of valves, tees and elbows in discharge line.

f) Additional equipment on discharge side of pump.

5. **Drive Data**

   a) Electric motor or other drive furnished by plant or installer.

   b) Horsepower, rpm, voltage, cycle, phase, AC or DC.

   c) Motor type (open, splash proof; totally enclosed, etc.)

   d) Who supplies pump base, pump coupling?

This topic discusses the characteristics of materials to be pumped. Subsequent topics will discuss the pumping system and its components so that the information requirements can be determined.

CHARACTERISTICS TO BE CONSIDERED

The first consideration in any pump application problem must be given to the characteristics of the material that is to be pumped. These characteristics will, to a large extent, determine the type of pump selected as well as the design of the piping system.

A detailed listing of the characteristics of the material to be pumped is the first step in pump selection. The most important characteristics are:

1. The chemical nature of the material pumped. This includes such items as acidity or alkalinity, corrosive nature, crystallizing or polymerizing tendencies, dissolved oxygen concentration. It is a good idea to make note of any historic data acquired during previous handling of material.

2. The viscosity, (resistance to flow), of the material at pumping temperature and ambient temperature. A substantial viscosity increase at ambient temperature could indicate that the material is unsuited for centrifugal pumps.

3. The specific gravity, (ratio of the weight of the fluid compared to the weight of an equal volume of water), of the material at pumping temperature and ambient temperature.

4. The pumping temperature during normal operation. The range over which the temperature of the material could vary during pumping operations may be an important factor. The vapour pressure (the pressure at which the fluid becomes vapour) of the fluid measured at the pumping temperature is also an important characteristic.

5. A number of other items indicating the nature of the material. These include toxicity, explosive nature, entrained gases, and the presence of abrasive particles. If there are over 3% solids by volume the fluid may qualify as a slurry and require special pump design.

For this topic, we will concentrate on problems created by materials pumped in water supply and wastewater operations.

WATER PUMPING

Water pumping systems are the most common of all pumping systems in plant operations. Water is a homogeneous and relatively non-corrosive liquid and presents few problems to a pumping system. To transport water, a wide range of pump types including positive displacement or centrifugal pumps can be selected.

Water must often be pumped in great quantities. When a centrifugal pump is operating at high capacity, low pressures may develop at the impeller eye or vane tips. If the pressure in these regions falls below the water vapour pressure, bubbles of vapour may form in the water. As the water moves to a region of higher pressure the bubbles will collapse. This formation and collapse of vapour bubbles is called **cavitation**. Because the bubble collapse is so sudden, the water will hit the impeller vane with such extreme force that it is likely to gouge out small pieces of the impeller. In addition to this pitting of the impeller, noise and vibration will be created. Cavitation may be reduced or eliminated by reducing the pumping rate. If cavitation is not remedied, serious mechanical damage to the pump will result.

Suspended or dissolved material in the water may cause problems. Sand grains suspended in the water pumped from a well could cause abrasive damage to the pump. Submersible pumps are most susceptible to this kind of wear. The running clearance of submersible pumps is normally 0.004 inches. Sand particles in this size range can cause abrasive damage by scoring the bearing surfaces. A simple test to determine if there is a sand problem can be conducted by filling a chemical (32 oz.) sample bottle with the water pumped and allowing the sediment to settle. If the sand covers the bottom of the bottle to a depth of one grain thick then the well pump is pumping water with more than the recommended concentration of sand. The only means of reducing sand being pumped from a well is by reducing the pumping rate. However, in doing this the operator must ensure that the discharge velocity of the water is high enough to keep the remaining sand in suspension all the way to the surface so that it does not settle out in the well.

Chemicals dissolved in water, such as chlorine and fluorides, make the water more corrosive and over long periods of time may cause corrosive damage to pumps. This potential problem should be given particular attention in water pumping systems where the chemical is added just before the water pump and the pump itself is used for mixing the chemical with the water. In this situation parts of the water pump exposed to higher than normal concentrations of the chemical may be subject to corrosive attack.

SEWAGE PUMPING

Generally, sewage is not pumped over great distances or at high pressures. The low head requirements of most sewage pumping systems as well as the pumps ability to pass small solid objects make the centrifugal pump better suited for this application. However, special consideration must be given to the variable nature of sewage in designing and operating a sewage pumping system. Unlike water, sewage is a heterogeneous mixture that may contain trash, rags, twigs, stones, etc. One must ensure that sewage pump will not be damaged by these objects. Perhaps a bar screen ahead of the pump will be required to remove the large objects. It is important that these bar screens be cleaned regularly. The impeller of a Centrifugal sewage pump must be of a special design with large openings to allow solid objects and stringy materials to pass through the pump without jamming it. Sewage, especially sewage containing storm water from a codlined sewer system, contains large amounts of sand and grit. As discussed earlier, this grit can cause abrasive damage to pumps. Most sewage pumps are designed with greater tolerance (i.e. greater amounts of clearance) and constructed with abrasion resistant alloys. Sewage may from time to time contain high concentrations of very corrosive industrial wastes. Frequently in sewage pumping systems the pump and perhaps the pump motor too (e.g. submersible sewage pump) will be completely immersed in the sewage. For this reason, the pump impeller, pump casing and any supporting structure should also be constructed of corrosion resistant alloys. Strict enforcement of the Sewer Use By-Law may protect sewage systems by reducing the concentrations of corrosive industrial wastes discharged to the sewers.

SLUDGE PUMPING

Sludge pumping systems have long been familiar to water pollution control plant operators. With the possibility of requirements for treating water treatment wastes, water supply plant operators will become more involved with sludge pumping. These sludges vary greatly in solids content but generally are of higher viscosity; that is, they are fairly thick and have more resistance to flow than water, but are still fluid when pumped.

Some sludges may contain coagulation chemicals or polymers, which will further increase the viscosity of the sludge. Basically, the pumps used for sludge pumping applications are of the positive displacement variety including both rotary and reciprocating types.

Reciprocating pumps are particularly well suited for pumping viscous fluids. This type of pump adds energy to a fluid system by means of a piston acting against a confined fluid. The viscous fluid coating the cylinder walls serves as an additional "packing". Rotary pumps are also well suited for high viscosity fluids. This type of pump traps a quantity of fluid between rotating gears or lobes and moves it along towards the discharge point. Some lubricating action by the viscous fluid decreases wear of the pump parts.

The positive displacement type sludge pump must be protected from excessive quantities of abrasive grit that might accumulate in the sludge. For this reason, the grit removal facilities that are included at the influent of water pollution control plants should be operated in the most efficient manner.

Centrifugal pumps are less effective in pumping viscous fluids. The centrifugal pump is so called because the pressure increase in the fluid generated by the pump is due largely to centrifugal action. During flow through the rotating impeller the fluid receives energy from the vanes, resulting in an increase in both pressure head and velocity head. However, since a large part of the energy of the fluid leaving the impeller is velocity head, it is necessary to reduce the velocity of the fluid and transform the velocity head into pressure head. This is accomplished in the spiral shaped volute casing surrounding the impeller which provides a gradually enlarging passage to effect a gradual reduction in velocity and increase in pressure. The difficulty in imparting velocity head to a slow flowing viscous fluid reduces the efficiency of centrifugal pumps for this application. When centrifugal pumps are used for higher viscosity fluids, there is a marked decrease in pump efficiency and capacity and an increased need for more horsepower.

CHEMICAL PUMPING

In water supply and water pollution control operations there are a wide variety of chemical pumping systems. As discussed earlier, it is necessary to consider the nature of these chemicals before pump selection. Pumps can be constructed of cast iron, cast steel, fabricated steel, bronze, brass and other alloys, moulded rubber, fiberglass or any other material that will be compatible with the chemical being pumped. Pumps handling corrosive chemicals require non-corrosive construction materials or internal protective coatings that the chemical will not attack. Special sealing devices are also necessary.

Chemicals used in water supply and water pollution control operations such as alum, ferric chloride, hypochlorite solution, fluoride solution, polymers, etc., must be fed to the water or sewage in accurate dosages. This requires a pump of very special design and construction called a metering pump. Metering pumps are usually positive displacement pumps generally classified as two main types; plunger (or piston) and diaphragm. The prime feature of these positive displacement pumps is that a definite quantity of chemical will be delivered for each stroke of the piston or diaphragm. The quantity of chemical will depend only upon the volume of the cylinder and the number of times the piston or diaphragm moves through the cylinder. On the other hand a centrifugal pump operating at a constant speed can deliver a variable volume of fluid depending on the head. Most metering pumps are designed with close tolerances (i.e. minimum amounts of clearance) between the pump casing and the internal parts. To maintain these close tolerances, it is sometimes necessary to strain or filter the chemical being pumped beforehand. If the chemical pumped is not strained, one must ensure that it is of high quality and does not contain particles or crystals that could damage the metering pump.

The specific gravity of the chemical will have to be considered in selecting the pump head. Equal pressures created by two different fluids require different heads. A ferric chloride solution can have a specific gravity of up to 1.5 (i.e. the ferric chloride solution can be one and one-half times heavier than water). Therefore the pump selected to pump ferric chloride solution to a certain height would have to generate approximately one and one-half times as much head as a water pump in the same situation.

Because of their precision construction, metering pumps will not tolerate extreme changes in the temperature of the chemicals they handle. Chemicals pumped at low temperature will show different pump capacity than the same chemicals at high temperatures. As the temperature of the chemical being pumped changes, the setting of the metering pump will have to be checked and adjusted if necessary to ensure proper dosage. In addition, the pumping temperature of certain chemicals like ferric chloride and alum must be carefully maintained above specified limits (45°F or 7°C) to prevent crystallization of the chemical which could cause damage to the metering pump.

SUBJECT:

PREVENTIVE MAINTENANCE

TOPIC: 7

PUMPING SYSTEM
CHARACTERISTICS

OBJECTIVES:

The trainee will be able to:

1. Define the various terms which apply to the pumping of liquid.

2. Explain the relationship between values expressed in pressure per square inch and that in feet of head.

3. Name the various components that make up the system head.

4. Explain the relationship between the variation of speed or impeller diameter and that of capacity, the pump head and horsepower.

5. Discuss the importance of the Net Positive Suction Head in any pumping installation.

6. Explain the difference between installations with suction head and suction lift.

7. Discuss the important factors involved in parallel operation.

PUMPING SYSTEM CHARACTERISTICS

BASIC THEORY

General

Pumping involves the addition of kinetic and potential energy to a fluid for the purpose of moving it from one point to another. The amount of energy imparted to the liquid depends on the characteristics of the pump. This addition of energy to the liquid will cause it to do work such as flow through a pipe or rise to a higher level. In any system of piping-process installation there is an inherent natural opposition to flow or "system head", the various vertical heights through which the fluid must be lifted as well as the friction losses occurring in the components of the system as a result of flow.

In effect, centrifugal pumps operate on the same basis as a boy swinging a pail full of water in a circle. Put a hole in the bottom of the pail and a stream of water is projected from the bottom. The faster the movement of the bucket the further the water will carry; the wider the hole, the more water that will flow through.

Converting this into centrifugal pump terms, the head developed by the pump will vary directly with outside diameter of the impeller. Similarly, sump capacity will vary with the size of the inlet opening and water passages. Thus, two pumps with the same diameter impeller could have widely different capacities, while still maintaining virtually the same pumping pressure or head. On the other hand, since head varies directly with the peripheral speed of the impeller, it is possible for two different sized impellers to develop identical heads, by running the smaller impeller faster to develop the same peripheral speed. This third factor is particularly important. For example, if a pump requires 10 hp at a given speed, doubling the speed of the pump would boost horsepower requirements to 80 hp. This is obviously impractical.

Capacity

Capacity is the quantity of water a pump will discharge against a given head. It is determined by the area through which flow occurs and by the velocity of flow. Velocity is dependent upon the peripheral speed of the impeller so that quantity is determined by the width of the impeller with constant speed. Of two impellers of the same diameter running at the same speed, the one having the greater width will deliver the greater capacity.

In specifying the pump capacity required for an installation, it should be stated in gallons per minute at the pumping temperature and any desired or imposed variation in the range of capacities should be clearly stated. Centrifugal pumps do not permit as much flexibility in capacity variations, without affecting pump efficiency, as does the reciprocating pump or other forms of displacement pumps. As a result, it is generally preferable to have the maximum efficiency of the pump occur at or near the normal capacity conditions.

Pressure

Pressures and their corresponding heads can be expressed either in absolute or gage units, such as 100 psi or 114.7 Psi abs. Gage pressure is merely that measured above or below the atmospheric pressure while absolute pressure is gage pressure plus the existing atmospheric pressure.

The properties of water are not appreciably affected by pressure in the normal range. Gage pressure is therefore commonly employed in problems dealing with liquids except where suction conditions must be dealt with. Keep in mind that a pump does not draw water into its suction; rather, the absolute pressure pushes the water into the pump. In most applications it is feasible to work in terms of gage pressure but, occasionally, a complicated problem can be clarified by working entirely in absolute values of pressure.

Practically any unit could be used to describe pressure; however, feet of water and pounds per square inch are standard in the hydraulics industry. Pounds per square inch is the weight on one square inch. Feet of water is the height of water that would give the equivalent weight on a square inch. Since most hydraulic calculations are done in feet of water, this is the term generally used to describe pump performance. It would take 3.21 feet of water to give a pressure of 1 psi.

If the liquid is other than water, its specific gravity in reference to cold water is generally known or can be determined and the column of liquid equivalent to 1 lb. per square inch pressure can be calculated by dividing 2.31 by the specific gravity of the liquid.

$$\text{Pressure (psi)} = \frac{\text{Head (feet)} \times \text{Specific Gravity}}{2.31}$$

$$\text{Head (feet)} = \frac{\text{Pressure (psi)} \times 2.31}{\text{Specific Gravity}}$$

Specific gravity has an effect on the height of a column of various liquids for equal pressures. Thus a pump which must handle 1.2 specific gravity brine against 100 psi net pressure would in effect raise a brine column 193 feet. If the pump has to handle water against the same net pressure it would in effect raise the water column 231 feet, while a pump handling 0.75 specific gravity gasoline against the same net 100 psi pressure would raise the gasoline column 308 feet.

It is obvious that if the pump designed to handle water was applied on brine service it would develop 231 feet head of brine or 120 psi pressure while if it was applied to pump 0.75 specific gravity gasoline it would develop 231 feet head of gasoline or 75 psi pressure.

Vapour Pressure

The vapour pressure of a liquid at any given temperature is that pressure at which it will flash into vapour if heat is added to the liquid, or conversely, that pressure at which vapour at the given temperature will condense into liquid if heat is subtracted.

For water and other homogeneous liquids the vapour pressure has a definite value at any given temperature. Tables are available which give the vapour pressure of such liquids over a wide range of temperatures. Certain mixed liquids however such as gasoline are made up of several components each having its own vapour pressure and partial vapourization may take place at various pressures and temperatures.

If the pressure in a pumping system is not equal to or greater than the vapour pressure of the liquid, the liquid will flash into a gas. Thus, pressure must be available on the suction side of the pump when handling hot water or volatile liquids.

Suction Limitations

The importance of keeping within the suction limitations of any pump (centrifugal, rotary, piston) cannot be emphasized too greatly. A pump, by creating a vacuum at the suction (impeller eye on a centrifugal) utilizes atmospheric pressure (14.7 P.S.I. at sea level) to push the liquid into the pump. Because of this, the suction lift is limited theoretically to 33.9 ft of water maximum (14.7 P.S.I. x 2.31 x SG (1.0) = 33.9' water). Internal pump losses reduce this limitation even more. The dynamic suction lift should be calculated carefully at the required capacity to make sure that it is within the pump's capabilities. Even systems taking suction from a source above the pump can cause trouble when friction losses are too great. Always keep the pump as close to the liquid source as possible. Many pump performance curves will show the maximum practical dynamic suction lifts for a given pump or for given capacities from the same pump. Since the limitation is based on internal pump losses also, it can be seen that in any given pump the recommended suction lift is reduced as flow increases.

Before discussing various considerations related to pumping systems it should be noted that:

1. capacity varies as the speed or impeller diameter
2. head varies as the square of the speed or impeller diameter
3. horsepower varies as the cube of the speed or impeller diameter
4. heads can be measured in various units, such as feet of liquid, pounds per square inch pressure, inches of mercury and other depending upon the field and the units of measurement of the country
5. pressures and head readings can be in gage or absolute units
6. the difference between gage and absolute unit is affected by the existing atmospheric pressure and thus by the altitude
7. in handling liquids, the pressure at any point in the system can never be reduced below the vapour pressure of liquid

HEAD

The term "head" by itself is rather misleading. It is commonly taken to mean the difference in elevation between the suction level and the discharge level of the liquid being pumped. Although this is partially correct, this interpretation does not take into account all of the conditions that should be included to give an accurate description. In any pumping system the liquid must be moved through pipes or conduits which offer certain resistances or, in other words, have certain frictional losses. This energy dissipation, or head loss, is called a friction head while that energy which has been converted into velocity energy is called velocity head. Thus static heads, pressure heads, friction heads, and velocity heads may all be encountered. On the other hand, when considering a pump by itself, "head" is a measure of the total energy imparted to the liquid at a certain operating speed and capacity.

NET POSITIVE SUCTION HEAD (NPSH)

The Net Positive Suction Head (NPSH) is the amount of energy in the liquid at the pump centre-line. To have meaning, it must be defined as either available or required NPSH.

Required NPSH is the energy needed for a pump to operate satisfactorily; that is, to fill the pump on the suction side and overcome friction and flow losses from the suction connection to the point in the pump where more energy is added. In a centrifugal pump, NPSHR is the amount of energy (in feet of liquid) required to overcome friction losses from the suction opening to the impeller vanes, and to create the desired velocity of flow into the vanes. NPSHR is a characteristic of the pump. It varies with pump design, pump size, and operating conditions. The figure is determined by test or computation, and is supplied by the pump manufacturer.

Available NPSH (NPSHA) is the inherent energy in a liquid at the suction connection of the pump (regardless of the type of pump), over and above energy in the liquid due to its vapour pressure. NPSHA is a characteristic of the system and one you must know to properly apply a pump. It can be calculated or can be obtained for an existing system by taking test readings at the suction side of the pump.

Since there are also internal pump losses, the available NPSH in a system must exceed the pump required NPSH or reduction in capacity and efficiency, noise vibration and cavitation will result.

To calculate the Available NPSH:

NPSH (Available) = (Sum of Positive Factors) - (Sum of Negative Factors)

Positive Factors
- static suction head (if any)
- atmospheric pressure (if open or vented tank or sump)
- positive pressure (evaluate if closed pressurized tank at suction)

Negative Factors
- vapour pressure
- all friction losses
- static suction lift

All terms must be in feet taking into consideration the specific gravity of the liquid being pumped.

The purpose of NPSH ratings and calculations is to ensure that there is sufficient pressure on the fluid being fed to the pump so that the liquid does not boil in the suction system.

SYSTEM HEAD

One of the most important concepts to be understood in the correct application of centrifugal pumps is that of "system head", the natural opposition to flow inherent in any piping-process installation. In feeding into a network of piping, a pump works against a head imposed upon it by that network. The overall net-work is called, for our purposes, the "system" and it includes the pipe, fittings, valves, tanks and other components which may be in the flow path.

The total head of a system against which a pump operates is made up of the following components:

1. Static head
2. Difference in pressure existing on the liquid
3. Friction Head
4. Entrance and exit head losses
5. Velocity head

Static Head

To illustrate, consider a condition where we want to bring water at a certain rate from a source of lower elevation to some reservoir or water tank at a distant site and at a relatively much high elevation. To do this, in the most practical and economical way, a pump is used to move the water from the source through the pipe and eventually into the reservoir or tank. Before the size or type of pump can be determined, one must find out how much energy is required by the water to move it to the elevated tank; or what are the obstacles or elements against moving the water from one point to another.

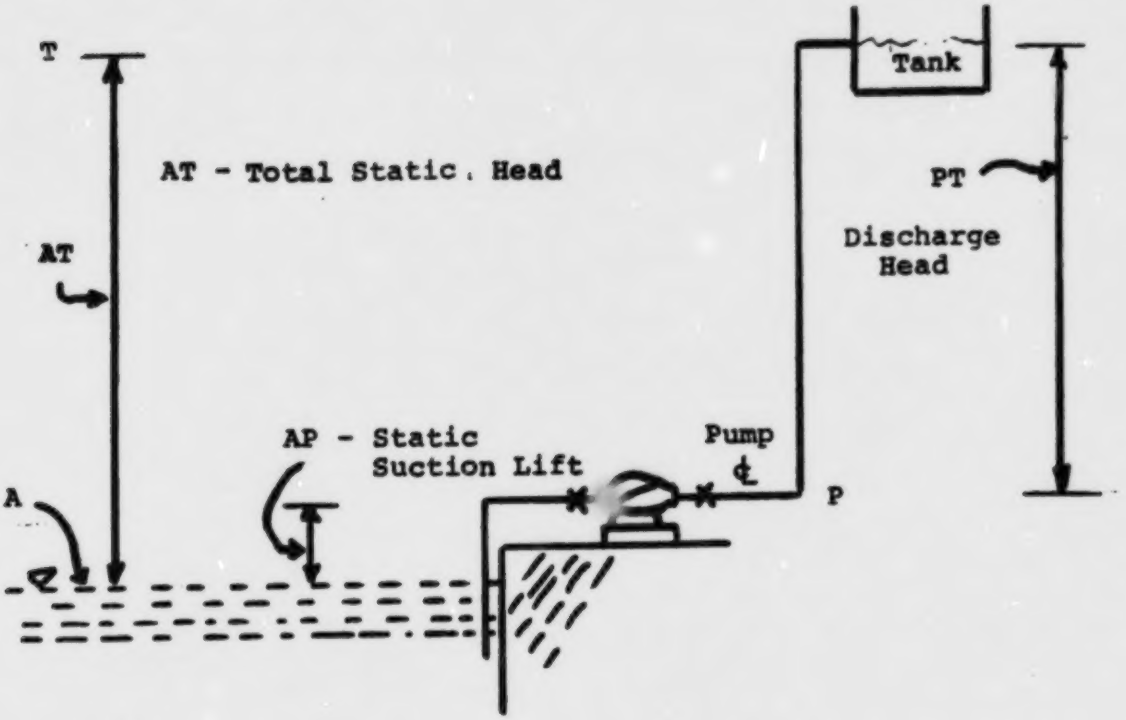


Figure 7-1 STATIC HEADS
(Suction Level Below Pump c/l)

In Figure 7-1, water is pumped from source A to an elevated tank. The vertical distance, which the water has to travel, measured from the free top water level at A to the highest point of free discharge at tank is called the total static head. In this particular case, it is represented by distance AT. Here it should be noted that the total static head is made up of the static lift, which is represented by distance AP and static discharge head by PT.

If the source of supply is above the centre line (c/l) of the pump the "static suction lift" is commonly referred to as the "static suction head". See Figure 7-2.

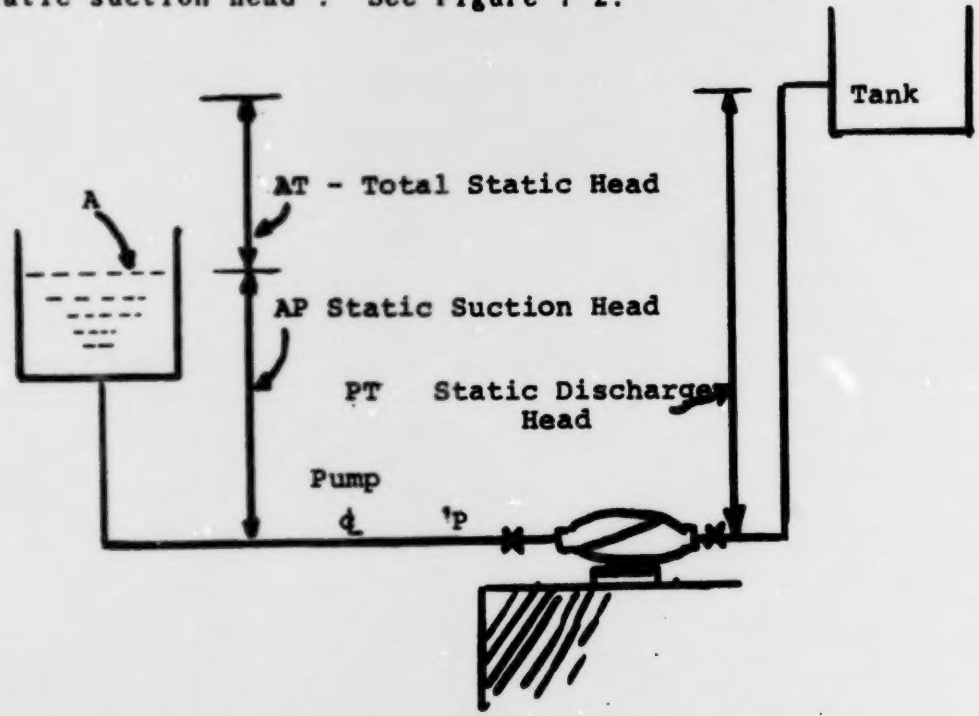


Figure 7-2 STATIC HEADS
(Suction Level Above Pump c/l)

Difference in Pressure

If either the suction or discharge liquid level is under a pressure other than atmospheric, practice varies as to whether this pressure should be considered at part of the static head or considered separately. The latter generally permits a clearer picture of the system.

Friction Head

Another element to overcome is friction, which is a kind of force which tends to slow down a body in motion. In this system, that force takes the form of a resistance to flow of water in pipes and fittings. That force or pressure required to overcome such resistance is called the "friction" head". Its magnitude in terms of feet of the liquid being moved varies as the amount or rate of flow of the liquid.

Whether or not it can be ignored depends upon the accuracy wanted and upon the accuracy of the pressure readings that can be made.

Prepared tables are available giving friction heads for various rates of flow through various sizes and types of pipe, expressed as feet of head per hundred feet of pipe. Friction loss factors are also given for various pipe fittings expressing the losses as equivalent to losses in specific length of straight pipe of the same size, or as percentage of velocity head.

The fact that pipes are made only in standard sizes materially affects design. Frequently, accurate design calls for in-between sizes, such as 1 3/4", but since these are not available, standard sizes must be used. Since it is a usual safety measure to use an oversized rather than an undersized pipe, friction losses through the system will be less than calculated.

There is a second reason for reduction in friction losses. The design of a system requires that the friction losses through the fittings, as well as through straight pipe, must be taken into account. The usual procedure is to add a percentage to the measure pipe length. This percentage varies considerably, depending upon the size of the installation, but experience indicates that the addition for the fitting is usually too large. As a result, the system actually has less friction loss than calculated.

Since pumps are selected on the basis of the CALCULATED friction loss, it is obvious that oversized pipe and too great an allowance for friction through the fittings will result in a pump having more capacity than required. The net results of an over sized pump are generally undesirable. In the first place, more money than necessary is expended for a larger unit. Second, the attempt to assure safety by oversizing frequently causes noise trouble because of the resulting increase in water velocity.

Entrance and Exit Losses

Usually the supply of a pump comes from some form of reservoir or intake chamber. The point of connection of the suction pipe to the wall of the intake chamber, or the end of the suction pipe projecting into the intake chamber is called the entrance of the suction pipe. The friction loss at this point is called the "entrance loss". The magnitude of this loss depends upon the design of the pipe entrance, a well designed bell mouth resulting in the lowest possible loss. Likewise on the discharge side of the system where the discharge line terminates at some body of liquid, the end of the discharge pipe is called the exit. Generally the end is the same size as the rest of the discharge piping and the velocity head of the liquid is entirely lost. In some cases the end of the discharge piping is a long taper so that velocity can be gradually reduced thus reducing the "exit loss".

Some consider entrance and exit losses as part of the friction head while others consider them separately to ensure that they are not overlooked. Considering them separately has the advantage that it can be easily seen if either or both losses are excessive.

Velocity Head

This is the head of the liquid being pumped due to velocity or the equivalent distance in feet through which water would have to fall to acquire that velocity. Thus velocity head can be calculated as follows:

$$hv = V^2/2g$$

Where V = velocity of water through the pipe in feet per second

g = acceleration due to gravity (32.2 feet per second)

hv = velocity head in feet

Velocity head in high head pumps is relatively small while in low head pumps, it is relatively high.

System Head Curve

The system head curve graphically presents the relation-ship of conditions discussed under System Head. The head H is plotted against capacity Q to obtain the "system head curve" or the "system demand curve" as shown in Figure 7-3.

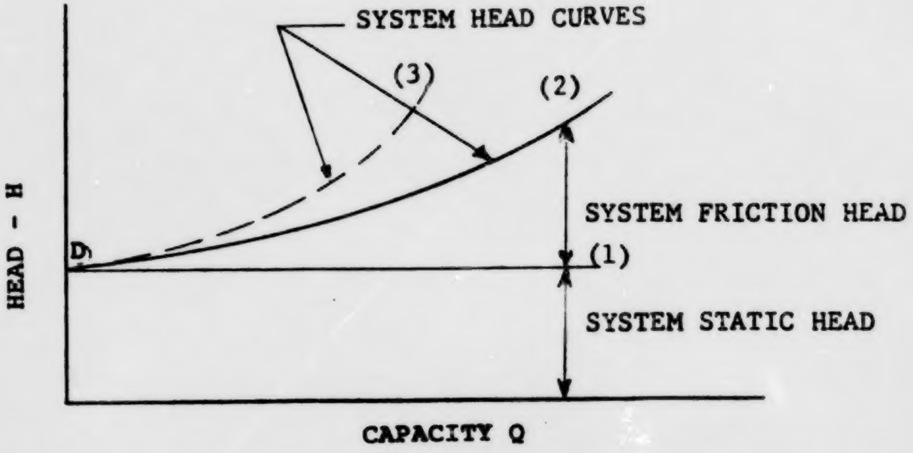


Figure 7-3 SYSTEM HEAD CURVE

If there were no friction losses along the piping, the system head curve would be represented by a horizontal straight line shown by line (1). With friction losses, the curve will be line (2). Now, if another piping system is used maybe with smaller pipes and fittings, the friction loss will increase for the same amount of flow as shown by the dotted line (3).

Note that the system head curve has a parabolic form since friction loss varies roughly as the square of the change in capacity. If the cross-sectional area of the pipe is assumed to be constant, friction loss varies roughly as the square of the change in liquid velocity through the pipe at the outset. It must be understood that any value assigned to friction loss can only be an approximation since friction losses will vary with size and type of pipe and fittings, with the interior condition of the pipe and with the character of the fluid being handled. It is usual to base the friction loss on constants that have been found from the average of pipe 10 or 15 years old, thus allowing for friction losses in excess of those that will be obtained when the pipe is new. As a result, the pump is generally designed for excess head and delivers over-capacity when installed in a new system or in one which has not suffered from pipe deterioration.

It will be noted that if there were no static head involved in the system, and there is only the friction head to work against, then point D in Figure 7-3 would shift down to zero, as shown in Figure 7 4.

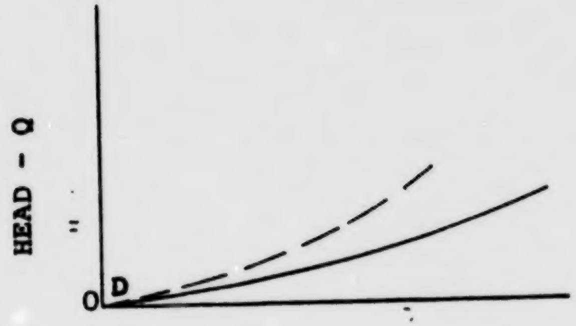


Figure 7-4 SYSTEM HEAD CURVE (No Static Head)

To plot a system head curve one must first establish the capacity desired and the head that the pump will work against. Let us assume that the existing pump is delivering 150 GPM at a 46 foot head.

Other flow/head relationship of this system can be calculated using the following formula:

$$\frac{\text{Original Head}}{\text{New Head}} = \frac{(\text{Original Flow})^2}{(\text{New Flow})}$$

By calculating the head for the following flows, a system head curve can be plotted as shown in Figure 2-5.

75 GPM, 100 GPM

125 GPM, 175 GPM

200 GPM

Thus at 75 GPM the head is calculated as follows:

$$\frac{46}{X} = \frac{(150)^2}{(\ 75)^2}$$

$$\frac{46}{X} = \frac{(2)^2}{(1)^2}$$

$$X = \frac{46}{4} = 11.5 \text{ feet head}$$

By similar calculation:

Flow		Head
100 GPM	=	20.5 feet
125 GPM	=	32.0 "
175 GPM	=	63.7 "
200 GPM	=	82.0 "

NOTE: In given system, the flow is doubled, the head increases to 4 times the initial value.

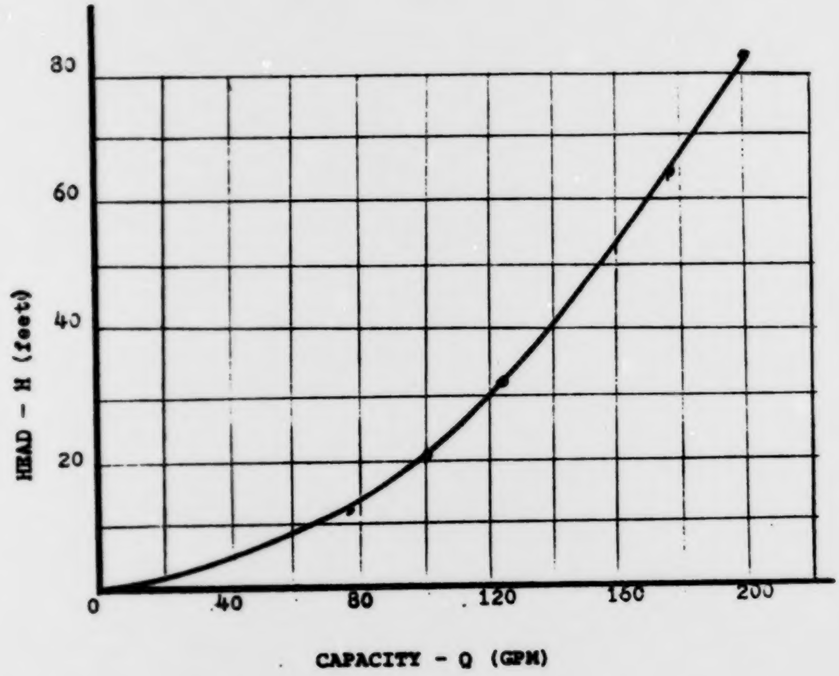


Figure 7-5 SYSTEM HEAD CURVE

Head Capacity Pump Curves

The manufacturer illustrates the performance of a pump by pump performance curves on a head capacity graph. The graph shows the interrelation of capacity, head, power and efficiency and is commonly known as the "characteristics of the pump".

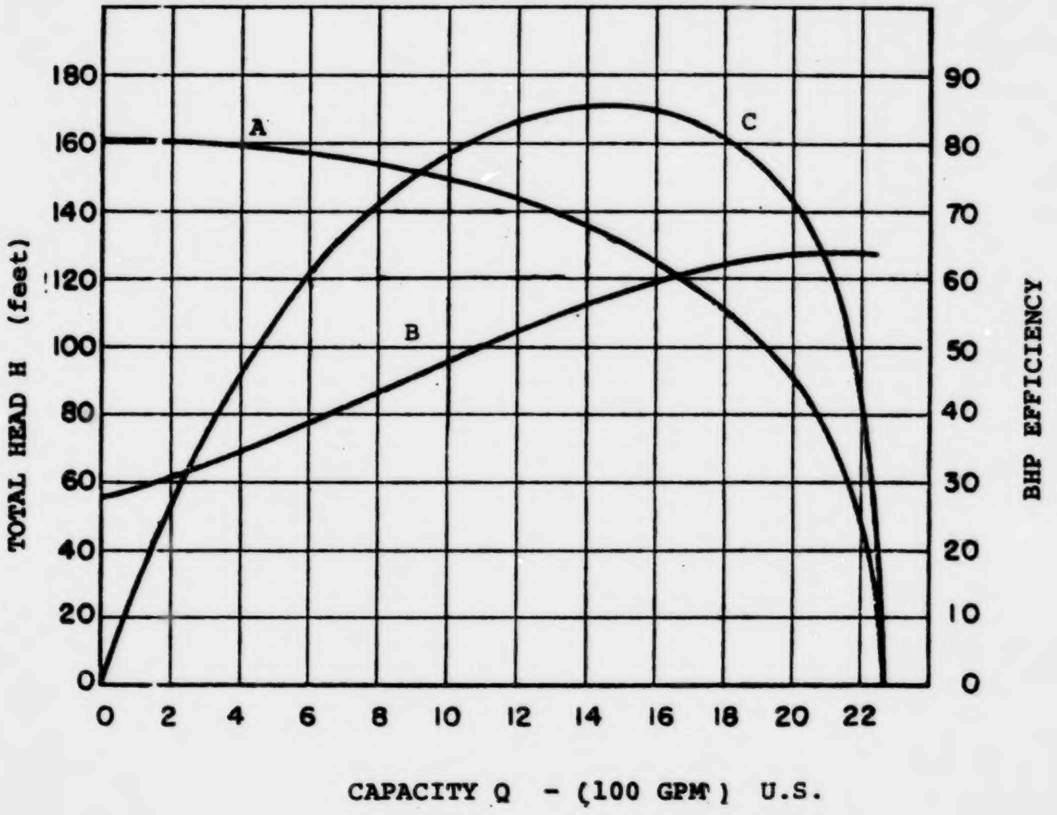


Figure 7-6 GRAPH-CHARACTERISTICS OF PUMPS

The line (A) sloping down from left to right represents the varying quantities of liquid delivered by the pump, with variations in head or pressure. The intersection of this line, with zero capacity line on the left shows the shut off head, or the head or pressure (160) developed by the pump when the discharge valve is shut.

The curve (B) showing the horsepower required to drive the pump slopes upward in the opposite direction, with the lowest point at the shut off of the pump.

These two lines show the complete performance of the pump for the one speed for which the curve is plotted. However, for convenience, another curve (C) showing the efficiency curve shows the amount of usable work done by the pump in percent of power delivered to the pump shaft.

Pump curves can be plotted to show variations with changes of speed as well as changes in head and capacities. However, most electric motors operate at constant speed, so that the usual practice is to show the pump characteristic for a single speed suitable for the conditions for which the pump is made. The changes in capacity, head, and horsepower, for various speeds, can be quickly calculated if desired.

USING THE SYSTEM HEAD CURVE

Always remember that the operating conditions for a pump can best be determined by plotting the system head curve and the pump characteristic curve to the same scale on one graph. The operating point will always be at the intersection of the two curves. All pump installations can be represented graphically as being a system head curve upon which the characteristic head-capacity curve of one or more pumps has been superimposed. Some systems are more complicated to calculate than others; these may require the calculation of individual portions of the system separately for later addition graphically to obtain the overall system head curve. Once the total head is known (and having considered all other factors), a pumping unit can be selected. A unit with a known head capacity characteristic curve is examined. The required capacity (Qa) and head (Ha) from the system head curve is compared with that on the head capacity characteristic curve by superimposing both curves on a graph as shown in Figure 7-7.

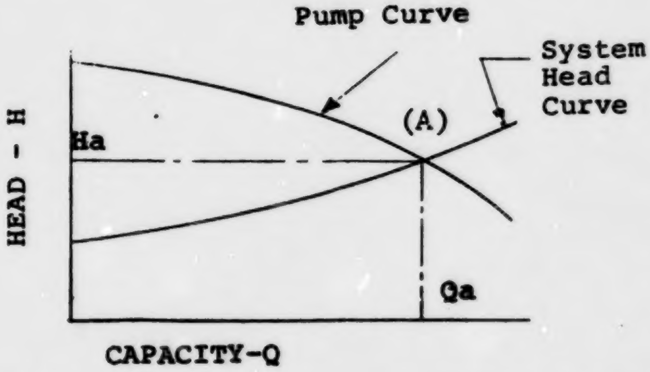


Figure 7-7 COMBINED PUMP/SYSTEM HEAD CURVES GRAPH

If the pump is rated at the capacity and head where the pump curve intersects the system head curve, then the pump matches the system. On the other hand, assume as shown in Figure 7-8 that the required capacity 150 GPM and head (46 ft) mentioned earlier is at a point just above the curve (A) of the pump being considered. Should pump A be used or the next larger size pump B?

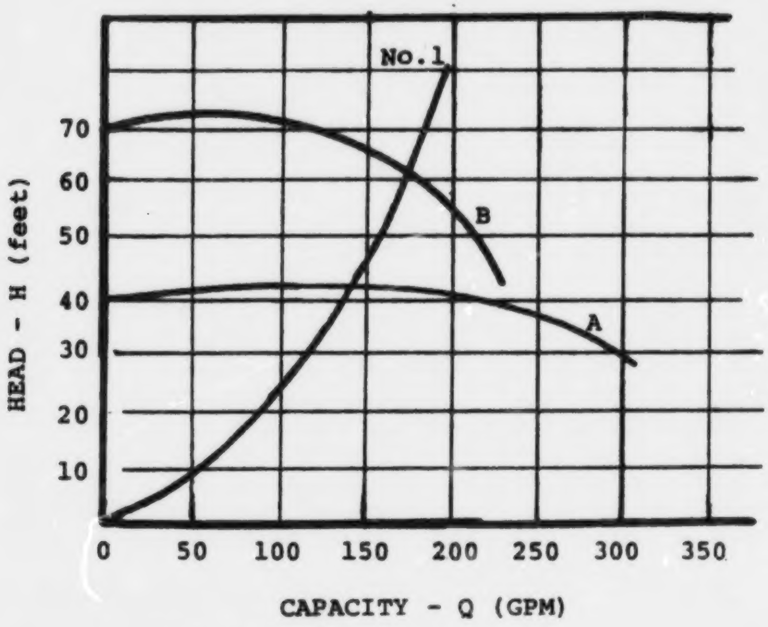


Figure 7-8 PUMP CAPACITY GRAPH

Line No. 1 representing the system head curve shown in Figure 7-3 is superimposed on the pump curve. It intersects the pump curve (A) at 140 GPM and 41.5 feet head and curve (B) where 175 GPM will be delivered at 62 feet.

Pump (A) could be safely selected because safety factors are deliberately included in the design of the pump.

One such inherent factor is in the carrying capacities assigned to pipe. Some tables provide such high factors of safety that the calculated friction loss at design flow may be too high by as much as 50%. Second, the allowance made for the resistance of the fittings is usually on the generous side. If accuracy is desired, a tedious series of calculations must be made, but as stated earlier, the designer usually bypasses this work by adding a percentage to the measured pipe length. Generally speaking, this method is satisfactory.

The selection of pump (A) can therefore be justified by both the conservative values given in pipe capacity tables and by the practice of making large friction allowances for the fittings.

The net results of selecting an oversized pump (B) are generally undesirable. In the first place, more money than necessary is expended for a larger unit. Second, the attempt to assure safety by oversizing frequently causes trouble because of the resulting increase in water velocity.

PARALLEL OPERATION

Since examples will lend further clarification, consider these typical common applications. Often, an installation will include problems of capacity variation over a wide range. This can be accomplished by placing pumps in parallel to discharge at the same pressure into the same header. Each pump can be placed in service or removed according to the system demand at the time. One approach would be the use of identical pumps placed in parallel. These would each develop the same head at any given capacity. When operating in parallel, the total capacity of two such pumps would be double the capacity of one pump at the head involved. This is illustrated graphically by Figure 7 9.

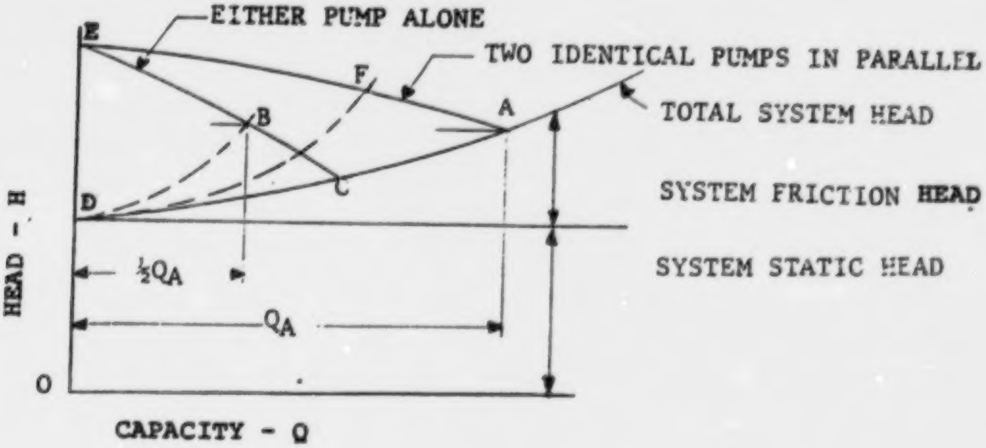


Figure 7-9 CAPACITY CURVE, TWO PUMPS IN PARALLEL

The two pumps together operate on curve EA intersecting the system head curve at point A. Note the combined operating curve starts at the no-flow or "shut-off" condition, E, of a single pump. Each succeeding point on the curve is developed by doubling the capacity of a single pump at any given head.

Were a single pump to operate alone at its design capacity, and therefore, at a head of C, its discharge valve could require throttling to create sufficient artificial friction loss to form a system curve DB. If the demand were to increase, the throttle must be made less severe until point C is reached. The system itself would limit the flow to this latter capacity. The determination of point C is important for with it, all possible operating points of a single pump have then been determined and the pump's driver can be sized to handle the maximum power requirement within the range EC.

Even if the pump has been factory tested and has demonstrated far greater carry out abilities and, depending upon the design, possibly greater power requirements, point C of the system head curve has determined the maximum power required for the pump when placed in the particular installation under consideration.

The development of the system head curve and a knowledge of demand variations to be expected can be of great assistance in determining the most economical division of the entire range into pump sizes needed. This can be done ideally when laying out a new system. For paralleling of the pumps to be accomplished properly, the units should be selected to have the same or nearly the same shut-off head. Otherwise, one pump having the higher shut-off head would tend to back the others off the line.

Such a system is represented by Figure 7-10. Pump #1 has the lower shut-off head and would operate at the shut-off condition at any capacity demand less than that of point B. The combined curve of the two pumps in parallel would be FBG. Were the capacity to decrease to B or below, pump #1 operates at the shut off condition. It could thus endanger itself if not shut down promptly, or if provision had not been made to by-pass adequate flow to carry away the heat that would be generated.

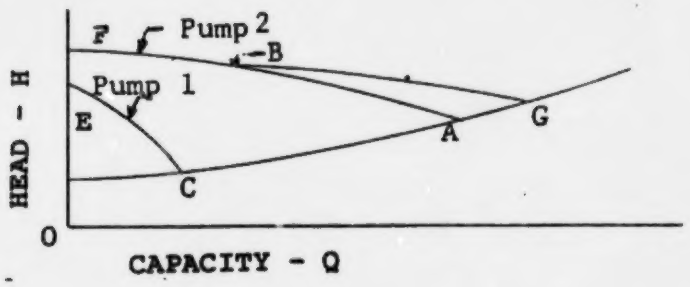


Figure 7-10 EFFECT OF OPERATING PUMPS WITH DIFFERENT HEADS IN PARALLEL

This arrangement is somewhat typical of municipal water works systems that have experienced and expansion of facilities due to population growth in outlying areas necessitating longer lines, often extensions of older mains now too small but too expensive to replace. The result is increased friction requiring additional pumps of larger capacity and higher head as well to fulfill the increased demand.

It should be noted further that, in this example, the larger pump would be required to operate over the range of low capacities where it would not be at its best efficiency point, nor possibly as efficient as a smaller pump designed for those capacities. A solution to this aspect of the problem, as well as the matter of paralleling properly, is often found in the purchase of a new impeller for the smaller pump, this impeller having a steeper head-capacity characteristic, or in other words, a higher shut-off head. Another possible solution may lie in stepping up the speed of the small unit, since head increases by the square of the change in speed of the pump. While a new and larger motor may be required, the cost will be easily justified in most cases, and will be considerably less than the purchase of a complete new pumping unit. In contemplating such a modification, full investigation of the particular pump's characteristics and design must be made with the manufacturer.

1. By the use of the system resistance curve, the pump performance can be checked against system requirements eliminating the usual trial and error adjustment.
2. Oversized pumps can cause excess velocity in the system and noise results.

SUBJECT: **TOPIC: 8**

PREVENTIVE MAINTENANCE **PUMP PACKING AND SEALS**

OBJECTIVES:

The trainee will be able to:

1. Recall the various types of compression packings along with the individual characteristics.
2. Recall the guides to packing selection.
3. Recall the procedure for installation of pump packing.
4. Recall the procedure for packing valves.
5. Recall the common packing failures.
6. Recall the mechanical seal:
    - (a) the components
    - (b) the types
    - (c) the factors governing selection
7. Recall the installation precautions for mechanical seals.

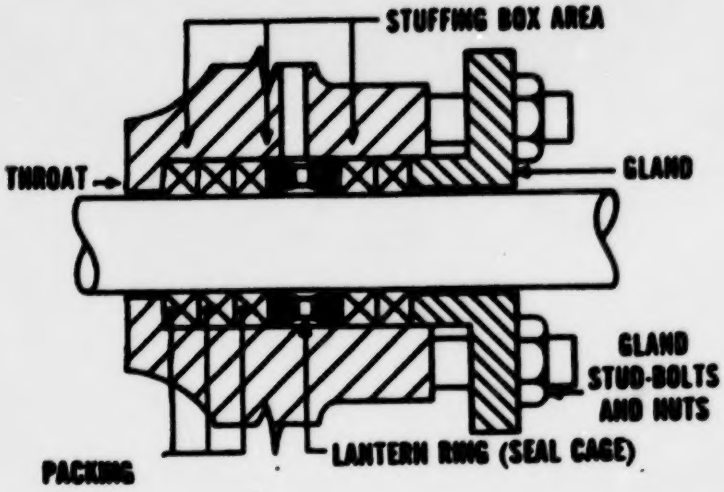
PUMP PACKING AND SEALS

The choice of the particular type of pump packing or seal is dictated by the manufacturer's engineering design personnel. This workshop will discuss compression type packings and mechanical seals.

COMPRESSION PACKINGS

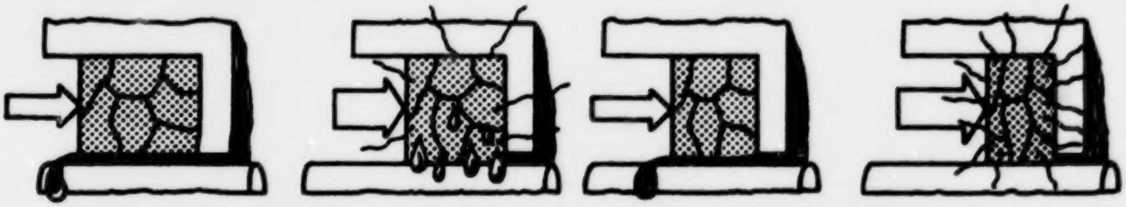
Compression packings are often referred to as soft packings or jam packings because they are packed into a stuffing box where their fit is regulated by compression from the gland or follower. A stuffing box (Figure 8-1) is the cylindrical cavity surrounding the point of shaft entry, which is fitted with a means of throttling leakage through it.

Packings are used as sealing elements in such equipment as centrifugal pumps, rotary pumps, reciprocating pumps and in industrial bronze, iron and steel valves as well as faucet valves. Where there is relatively no motion, slow or infrequent motion, such as in valves, packings can seal without leakage. Where rapid motion occurs, rotating or reciprocating, such as in centrifugal pumps, packings must permit leakage for lubrication, thus ensuring cool operation. Pump packings (Figure 8-2) have an impregnant built into them which acts as a primary lubricant for the start up and the break-in adjustment. During this period a certain amount of the primary lubricant is lost. Once the pump is on stream, external lubrication must be supplied to the packing to keep it running properly and to attain long life. If not, the primary lubricant in the packing will bleed out, friction will increase with the attendant generation of heat, the primary cause of packing failure. Compression packings are expendable materials which are expected to wear out in preference to shafts and shaft sleeves which are thus preserved.



TYPICAL
STUFFING BOX
ARRANGEMENTS AND
DESCRIPTION
OF PARTS

Figure 8-1 STUFFING BOX



New packing

No fluid leakage: saturant oozes out

Fluid lubricates

No saturant left

Figure 8-2 NEW PACKINGS ARE SATURATED WITH AN "EMERGENCY" LUBRICANT FOR INITIAL BREAK-IN

When the pump suction pressure is above atmospheric pressure the pumped liquid lubricates the packing, (Figure 8-3). If this pressure is below atmospheric pressure fluid may not reach the packing. Thus a lantern ring is employed and lubrication is supplied through a piping arrangement from the pump discharge pressure (Figure 8-4). These conditions are both employed when pumping clear liquids. When pumping slurries an external source of clear liquid which is at a higher pressure than the slurry must be force-fed into the lantern ring. Two methods of handling slurry, in current use, are shown in Figures 8-5a and 8-5b.

A valve, expansion joint, autoclave doors and other basically static applications are packed differently than a pump. In contrast to a pump packing, which must leak, a valve packing must not leak.

Compression packings used in valves, expansion joints, autoclaves and other static types of applications function like a gasket, as compared to a pump packing which must both seal to a degree and carry lubricant for rotating or reciprocating shafts. Valve stem packings must provide a dense structure that will not permit movement of the liquid or gas through the body of the packing itself.

For a shaft or rod in motion, be it rotating or reciprocating, choose a packing that has a primary lubricant in it.

Then, when the equipment is on stream, make sure the packing has supplemental lubrication from either the pumped fluid or an external fluid. If the application is static, such as valve stems, expansion joints, etc., make sure that packing has a stable lubricant in it that won't ooze out or dry out due to pressure/ temperature condition of the process fluid.

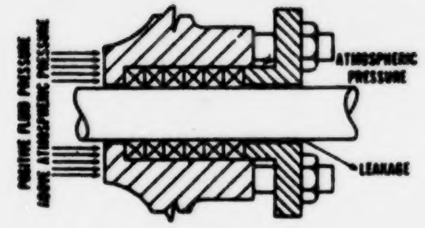


Figure 8-3

TYPICAL STUFFING BOX WHEN PUMP SUCTION PRESSURE IS ABOVE ATMOSPHERIC PRESSURE

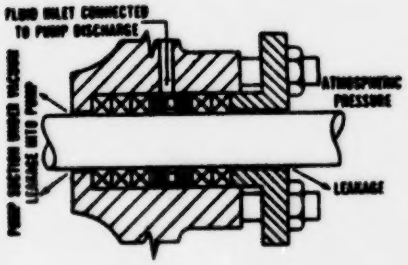


Figure 8-4

TYPICAL STUFFING BOX WHEN PUMP SUCTION PRESSURE IS BELOW ATMOSPHERIC PRESSURE

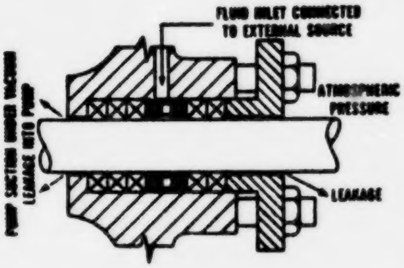


Figure 8-5

TYPICAL STUFFING BOX WHEN PUMP IS PUMPING SLURRIES.

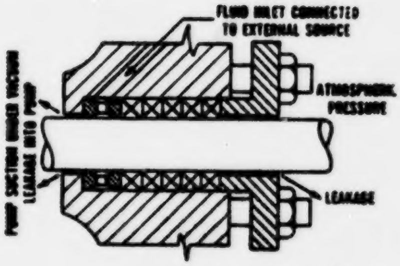


Figure 8-6

ANOTHER TYPE OF STUFFING BOX WHEN PUMP IS PUMPING SLURRIES.

TYPES OF COMPRESSION PACKING

Compression packing are made in a variety of shapes, sizes and constructions, from numerous types of materials. Certain types of these soft packings have evolved over the years that best suit specific applications. A knowledge of these types of packings, along with their equipment usage, will present misapplication and will reduce maintenance downtime.

Types of Braided Packings

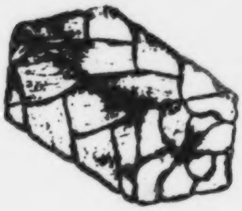
Engineers and maintenance men often refer to "braided" packings. This is not sufficient as there are several types of braided packings on the market.

The most popular braided packings used on rotating equipment are square interbraid, square plaited, braid-over-braid and braid-over-core types. Of these four, the square plaited are most popular. Each is made on a different type of braiding machine and has a certain amount of strength because of the braid pattern. The following shows the braid patterns and typical usage on rotating pumps.

1. Square Interbraid

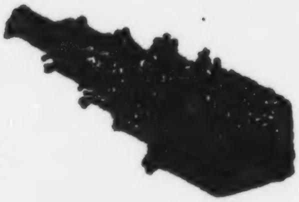
Actually formed square, with yarns diagonally braided, will retain square shape more than braid-over-braid.

2. **Square Brain (Plaited)**



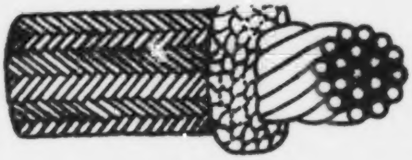
Soft, absorbent structure can carry high amount of lubricant.

3. **Braid-Over-Braid**



Braided round, then calendered square. Very dense because of individual jacket over-jacket construction.

4. **Braid-Over-Twisted Core**



Braided round, then calendered square. Can carry high amount of lubricant.

These four different types of braided structures can be made from a variety of materials. Currently special materials such as "Kevlar" type packing are gaining popularity.

Under certain conditions, such as mechanical vibration or excessive gland clearances, it may be necessary to use yarn reinforced with metal. Normally, copper or brass wire inserted yarns are used. When clearances at the bottom or top of the stuffing box are too great, extrusion of the packing can occur, and then lead or other metallic bull rings are used. Other materials used in packing include ranine, flax, asbestos, cotton, teflon filament yarns, graphite filament yarns, graphite filament yarns, and in some cases metal wire and metal ribbon.

Packing manufacturers impregnate the yarns and/or various braided structures with lubricants that will withstand temperature and washout from the pumped fluid, as well as provide lubrication for the break-in and start-up periods. Once on stream, the pump fluid must lubricate the packing.

Since packings are used against water, steam, oil, acids, caustics, gasolines, food, beverages, etc., it is only natural that the manufacturer must have several types of primary lubricants to resist pumped fluid wash-out.

Metallic Packings (Figure 8-6)

Metal packings of lead, copper, aluminum, phosphor bronze and nickel are now made either spirally wrapped or in a folded, twisted construction, often around a resilient or compressible core. Both types are ordinarily oiled and graphited, although they may be furnished dry for special applications. Endless spun and compressed lead rings form a superior condenser packing and also have other uses as gaskets, end rings, etc. One of the first flexible metallic rings was the spirally wrapped type which is still used for heavy pressure duty work. Metal and fabric packings are generally lead, copper or aluminum over braided or twisted asbestos or braided flax.

Metal and plastic packings normally have lead foil over a plastic core but may be a braided metal wire over plastic.

Metal and resilient elastomer-core packings such as channel packings combine metal and fabric on the face only. Metallic cores may be lead or copper. Lead is usually spirally wrapped, while copper is a braided wire.

Plastic Packings (Figure 8-7)

Lubricated plastic packings contain asbestos fiber, graphite or mica, oil or grease. Some contain metal particles.

Others also contain wood particles which form additional lubricant reservoirs. These packings are sold:

1) in bulk

2) in sticks for injection by heavy duty screw type guns

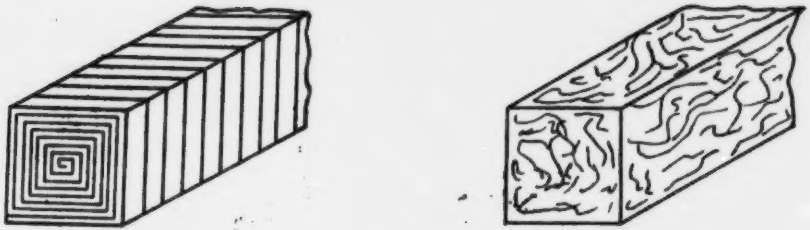
3) enclosed in a braided jacket

    a) open cotton thread for handling only

    b) asbestos jacket to help retain packing in a stuffing box

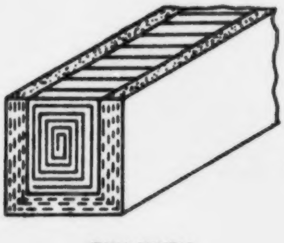
    c) short lengths in braided cotton tubing

4) die formed rings.



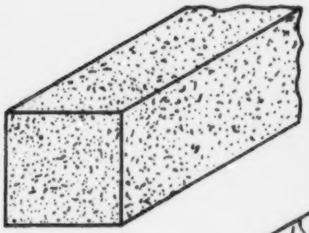
METALLIC
(SPIRALLY WRAPPED)

METALLIC
(TWISTED AND FOLDED)

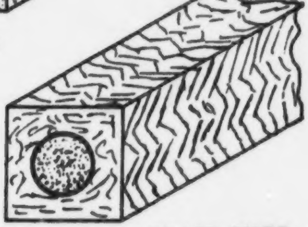


CHANNEL
(METALLIC FACE)

Figure 8-6 Metallic Packings



PLASTIC



BRAID OVER
PLASTIC CORE

Figure 8-7 PLASTIC PACKINGS

These packings are useful where packing must supply a part or all of the lubrication. For medium and high speed rotary service, additional lubrication or cooling is usually necessary.

The second type of plastic packing is the dry bonded type which has a composition similar to the lubricated type except that an elastomeric binder is used instead of a lubricant. Asbestos fiber and graphite or mica with or without metal particles, depending on the type and temperature of service, are bonded with natural rubber or Neoprene. Small amounts of lubricant are used, but function only as softeners and do not reach the shaft to serve as a lubricant. Like the lubricated types, these styles are available in different forms:

a) Bulk

b) Flat coils or spirals

Plastic packings can be supplied with a plain or wire reinforced asbestos jacket to enable the sealing of higher pressures with one style of packing.

Rubber and Duck Packings

Rubber and Duck types are made by laminating plies of cotton duck which has been treated with a rubber compound. They are normally press-cured in slab form and then cut in strips of a desired size. Then they are generally impregnated with oil and given a graphite finish. These packings are generally used on the liquid ends of reciprocating services, such as rams or pistons. The rubbers commonly used are natural, Neoprene, and SBR.

Asbestos Fabric Packings

Asbestos fabric constructions have been used on reciprocating pumps on the steam end. These packings may have a round core or rectangular back of rubber. The rubberized asbestos cloth is folded or rolled on itself or on the core or rubber back to the required size. It is then vulcanized, lubricated with vegetable oil and given a graphite finish.

Die Form Rings

Die formed rings and die formed combination sets of packing are often used because:

a) The rings are die formed to the exact size of the stuffing box.

b) The exact number of rings to fit a stuffing box eliminates guesswork.

c) No waste due to inaccurate cutting of rings from spools.

d) Repack stuffing boxes faster, reduces downtime.

Channel Packings

Another type would be a channel packing where a channel made either of asbestos or duck cloth with a metallic face of either lead or copper is used. This style is very effective on high pressure reciprocating applications for either steam or hydrocarbons.

Lubricants Used In Packings

To meet the chemical and mechanical conditions, packing manufacturers use both solid and fluid lubricants. Commonly used solid lubricants include TFE, graphite, mica and moly disulphide. Fluid lubricants include oil, greases and waxes of animal, petroleum, vegetable and synthetic types.

GUIDE TO PACKING SELECTION

Probably one of the most frustrating tasks of maintenance personnel is how to select a certain type of compression packing for a given application. The manufacturers' catalogues may list anywhere from a few styles up to 80 or 90 types. If some general basic understanding of compression packing selection is known, then selection becomes less complex.

Where conditions are not severe and pressures are low, a single type of soft plastic packing or fabric packing can be used successfully. But as pressures increase it is often advisable to include a more dense packing, such as a braided type of asbestos or metallic packing to support the soft plastic rings. One ring of these more dense materials would be used at the bottom of the box and another under the gland to prevent extrusion of the soft materials at these two points. For the same reason, if a lantern ring is used the dense packing is used on both sides of the lantern ring. As operating pressures increase, a greater number of these more dense rings are used in the stuffing box with a decreasing number of the softer, more sensitive packings.

With the large use of centrifugal and rotary pumps in industries today, packing selection for this equipment becomes of utmost importance. Tied in directly with this is the selection of packings for valves. Thus probably 75% of packing applications boil down to rotating pumps and valves.

To determine which packings should be used, there are certain basic questions that must be answered and these determinants are as follows:

1. Dimensions
2. Fluid or gas handled
3. Stuffing box temperature
4. Stuffing box pressure
5. Type of motion
6. Speed
7. Shaft or rod material

8. Cycles
9. Mechanical condition of equipment
(shaft, bore, etc.)
10. Special requirements.

Rather than attempt to make your own choice of packing material, it is advisable that the above information be made available to a packing supplier or manufacturer.

INSTALLATION OF PACKINGS

Packing the Pump Correctly

The importance of packing the pump correctly cannot be over-emphasized. Many packing failures are due to incorrect installation of the packing. There is one correct way to pack a pump which is as follows:

1. Remove all old packing from the stuffing box. Clean box and shaft thoroughly and examine shaft or sleeve to see if wear is excessive. Check bearing by lifting shaft up and down: do not expect packing to act as a bearing.
2. Use the right cross section of packing or preformed rings to be sure the packing will fit and can compensate for shaft wear, if any. To determine the correct packing size, measure the

diameter of the shaft (inside the stuffing box area if possible) and then measure the inside diameter of the stuffing box itself (to give the O.D. of the ring). Subtract the I.D. measurement (the shaft diameter) from the O.D. measurement and divide by two. The result is the required size. Standard compression packings are generally made up in increments of 1/16" from the smallest size available. For most applications, if the calculated size figures under 1/16", it is best to use the next size larger. Thus, if the shaft measures 2 1/2" and the inside of the stuffing box 3 7/16" (giving a packing space of 15/16"), the size of coil packing to use would be 1/2" rather than 7/16".

3. When using coil or spiral packing always cut the packing into separate rings. Never wind a coil of packing into a stuffing box. Rings can be cut either with butt (square) joint or diagonal joints depending on the method used for cutting.

The best way to cut packing rings is to cut them on a mandrel the same diameter as the shaft in the stuffing box area. If there is no shaft wear, rings can be cut on the shaft outside the stuffing box if there is room.

Hold the coil packing tightly and firmly on the mandrel but do not stretch excessively. Cut the ring and try it in the stuffing box to make certain it fills the packing space properly with no gap in the joint at the O.D. of the ring.

Each ring can be cut in the same manner or the first ring can be used as a master from which the balance of the rings are cut. If the butt rings are cut on a flat surface be certain that the side of the master rings and not the O.D. or I.D. surface is laid on the rings to be cut; this is necessary so that the slight bevel at the end of the rings be reproduced. Remember the O.D. circumference of the ring is greater than the I.D. circumference. When cutting diagonal joints use a maple mitre board so that each successive ring can be cut at the correct angle.

It is necessary that the rings be cut to the correct size, otherwise service life is reduced. This is where die formed rings are of great advantage as they give you the exact size ring for the I.D. of the shaft and O.D. of the stuffing box and there is no waste because of incorrectly cut rings.

4. To cut rings from a packing that is unusually soft or that has a tendency to fray excessively, it is advisable to apply a small piece of tape at the point where the cut is to be made, then cut through the tape. If the tape is thin it can be left on the ring.

5. Install one ring at a time. Make sure it is clean and has not picked up any dirt in handling. If clean oil is available, lubricate the shaft and the inside of the stuffing box. Seat rings firmly. Joints of successive rings should be staggered and kept at least 90° apart. Each individual ring should be firmly seated with a tamping tool. When enough rings have been individually seated so the nose of the gland will reach them, individual tamping should be supplemented by the gland. Never depend entirely on the gland to seat a set of rings properly, this practice will jam the last rings installed but leave the bottom rings loose in the box. The result is excessive and rapid wear of outer rings, erratic packing performance; sometimes

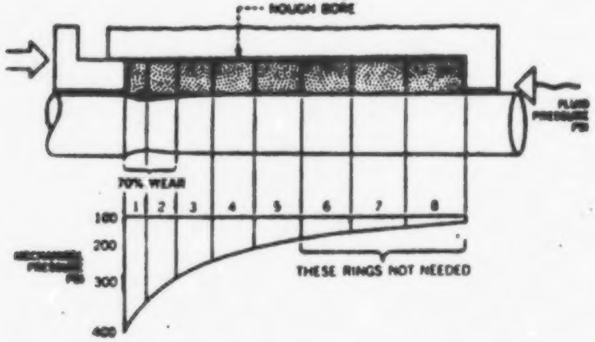
twisting and tearing of the bottom rings because they are loose in the stuffing box.

6. After the last ring is installed, take up bolts finger tight or very slightly snugged up. Do not jam the packing into place by excessive gland loading. Start pump, and take up bolts until leakage is decreased to a tolerable minimum. Make sure gland bolts are taken up evenly. Stopping leakage entering at this point will cause the packing to burn up.
7. Allow packing to leak freely when starting up a newly packed pump. Excessive leakage during the first hour of operation will result in a better packing job over a longer period of time. Take up gradually on the gland as the packing seats until leakage is reduced to a tolerable level. Never try to stop leakage entirely.
8. When specified by the pump manufacturer, provide means of lubricating the shaft and packing through a lantern ring by supplying water, oil, grease or liquid handled in the pump. Fittings for this purpose are standard on many pumps.

9. If the stuffing box has a lantern ring (see illustration), make sure the lantern ring, as installed, is slightly outboard of the fluid inlet so it will move in line with the inlet as follower pressure is applied.

10. Replace packing when leakage cannot be controlled by further take-up on the follower.

11. On both centrifugal and reciprocating pumps about 70° of wear is on the outer two packing rings nearest the gland. However, each additional ring does throttle some fluid pressure. And on most machines there must be enough rings so if one fails another does the sealing and the machine need not be shut down.

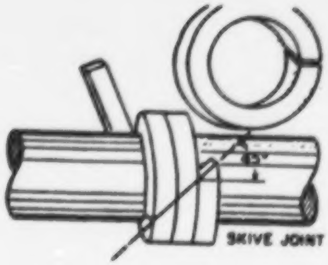


The mechanical pressure curve above shows eight packing rings only about five of which are needed, according to the pressure curve, as the bottom three do little sealing. The advantage of using fewer rings is less rod wear, and the stuffing box design is simpler, takes less material. But wear isn't the only problem. With high temperatures, high pressures, corrosive chemicals or abrasive particles in the fluid, more rings may be the only solution for some services. In such cases, the bottom ring contacting the casing fluid may take most of the beating from these severe service conditions.

Packing Valves Correctly

As with pump packing, the first step to getting the most out of a valve packing is correct installation. Here is the right way to pack valves:

1. Carefully perform all the operations outlined under steps 1, 2, 3, 4, and 5, as given for pumps and agitators. Rings used on valves and expansion joints are generally cut with a bevel joint (illustrated below).



In preparing bevel cut (45°) the first ring is carefully cut and then tested on the stem.

2. Bring the follower down on the packing to the point where heavy resistance to wrenching is felt. During this time turn valve stem back and forth to determine ease of turning. Do not torque down to the point where the stem won't turn.

TROUBLESHOOTING PACKING FAILURES

Packings may fail for a variety of reasons. Besides improper installation, packing failures are often due to worn or faulty equipment, shaft misalignment, shaft whip, bearing misalignment, uneven take-up on the gland bolts and other causes. If you are having trouble, carefully remove and examine the old packing set. Do not throw the set away because it often gives clues as to the condition of the equipment...and may be the means of solving the problem. The following clues and possible troubles were found by examining sets of packing which failed in service:

1. Clue: Excessive reduction in cross-section of packing directly beneath rod, shaft or plunger.

   Possible Trouble: Low bearings with shaft or rod riding packing In service early leakage occurs around the top of the shaft or rod.

2. Clue: Excessive reduction in thickness of packing directly over or on either side of rod or shaft.

   Possible Trouble: Rod or plunger out of alignment, and in the case of the rod or shaft, the bearings may be badly worn, causing whipping of the shaft.

3. Clue: Part of one or more rings missing from set.

   Possible Trouble: Bottom of stuffing box badly worn, with packing being extruded into system. Considerable difficulty, such as packing getting into valves, traps or contaminating fluid handled, may result.

4. **Clue:** Axial bulge in one or more rings.

Possible Trouble: Adjacent rings cut too short or too long, causing packing under pressure to be deformed.

5. **Clue:** Packings showing tendency to sheet out or extrude between rod or shaft and the gland follower.

Possible Trouble: Excessive gland bolt pressure. Too much clearance between the shaft and the gland follower.

6. **Clue:** Rings next to gland followers badly deformed with bottom rings in fair condition.

Possible Trouble: Improper installation of packings and excessive gland bolt pressure used.

7. **Clue:** Wearing face of rings dried and charred with rest of packing in good condition.

Possible Trouble: High temperatures and lack of adequate lubrication.

8. **Clue:** Innermost ring deteriorated.

Possible Trouble: Packing incompatible with fluid handled.

MECHANICAL SEALS

A mechanical seal is a sealing device which forms a running seal between rotating and stationary parts. The design of liquid handling equipment with rotating parts today would include the consideration for the use of mechanical seals. Advantages over conventional packing are as follows:

1. Reduced friction and power losses
2. Zero or limited leakage of product
3. Elimination of shaft or sleeve wear
4. Reduced maintenance

The wide variety of styles and designs together with extensive experience allows the use of seals on any sealing problem.

A mechanical seal must seal at three points:

1. Static seal between the stationary part and the housing.
2. Static seal between the rotary part and the shaft.
3. Dynamic seal between the rotating seal face and the stationary seal face.

Components of Mechanical Seal

Figure 8 8 shows a basic seal with these components.

1. Stationary seal part positioned in the housing with preload on the "O" ring to effect sealing and prevent rotation.
2. Rotating seal part positioned on the shaft by the "O" ring. The "O" ring seals between it and the shaft and provides resiliency.
3. The mating faces The faces are precisioned lapped for a flatness of 3 light bands and a surface finish of 5 microinches.
4. Spring Assembly Rotates with the shaft and provides pressure to keep the mating faces together during periods of shut down or lack of hydraulic pressure.
5. Driving member positions the spring assembly and the rotating face. It also provides the positive drive between shaft and the other rotating parts.
6. End movement - As wear takes place between the mating faces the rotating face must move along the shaft to maintain contact with stationary face. The "O" ring must be free to move.

A mechanical seal operates as each basic component performs its duty. Liquid pressure in the seal chamber forces the faces together and provides a thin film of lubricant between them. The faces, selected for low frictional qualities, are the only rubbing parts. These basic components are a part of every seal. The form, shape, style and design will vary greatly depending upon service and manufacture. The basic theory, however, remains the same.

Types of Mechanical Seals

Mechanical seals can be classified into the general types and arrangements shown below. Understanding these classes provides the first step in proper seal selection:

Single seals (inside, outside)

Double seals

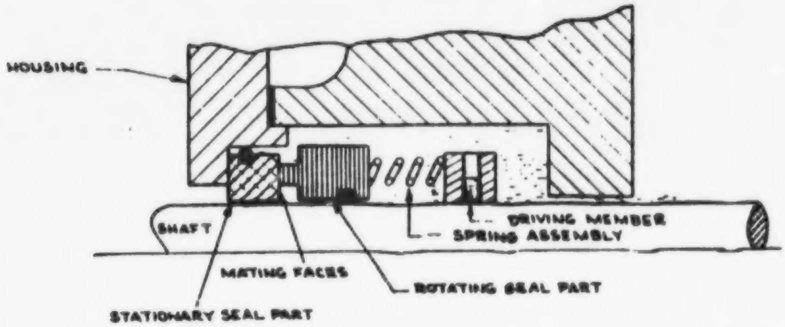


Figure 8-8 Basic Mechanical Seal

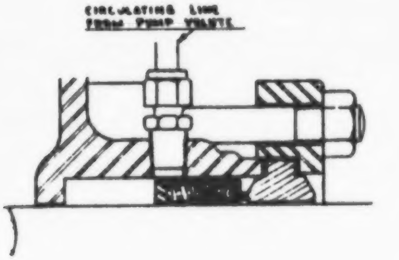


Figure 8-9 Single Seals
Inside Unbalanced
Unbalanced

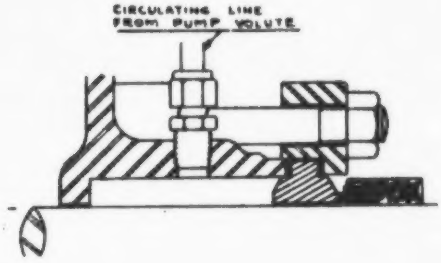


Figure 8-10 Single Seals
Outside
Unbalanced

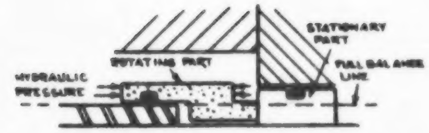


Figure 8-11 Single Seals
Balanced

Single Seals, Inside Unbalanced (Figure 8-9)

The single inside seal mounts on the shaft or sleeve within the stuffing box housing. The pumpage is in direct contact with all parts of the seal and provides the lubrication for the faces. The full force of pressure in the box acts on the faces providing good sealing to approximately 200 P.S.I.G. This is the most widely used type for services hand-ling clear liquids. A circulation or by-pass line connected from the volute to the stuffing box provides continual flushing of the seal chamber.

Single Seals, Outside Unbalanced (Figure 8-10)

This type mounts with the rotary part outside of the stuffing box. The springs and drive element are not in contact with the pumpage, thus reducing corrosion problems and preventing product accumulation in the springs. Pressures are limited to the spring rating, usually 35 P.S.I.G. Usually the same style seal can be mounted inside or outside. The outside seal is easier to install, adjust and maintain.

Single Seals, Balanced (Figure 8-11)

Balancing a seal varies the face loading exerted by the box pressure, thus extending the pressure limits of the seal. A balanced rotating part utilizes a stepped face and a sleeve. Balanced seals are used to pressures of 2000 P.S.I.G. Their use is also extensive on light hydrocarbons which tend to vaporize easily.

Balanced outside seals allow box pressure to be exerted toward the seal faces, thus allowing pressure ranges to above 150 P.S.I. as compared to the 35 P.S.I.G. limit for the unbalanced outside seal.

Double Seals

Double seals use two seals mounted back to back in the stuffing box. The box is pressurized with a clear liquid from an outside source. These seals are used extensively on slurry service.

Selection of Mechanical Seals

The proper selection of a mechanical seal can be made only if the full operating conditions are known. These conditions are as follows:

1. Liquid
2. Temperature
3. Characteristics of liquid
4. Pump model

Liquid

Identification of the exact liquid to be handled provides the first step in seal selection. The metal parts must be corrosion resistant. These parts usually available in steel, bronze, stainless steel, or Hastelloy provide a wide choice to meet specific services.

The mating faces must also resist both corrosion and wear. Carbon, glass-filled Teflon, Stellite or tungsten carbide allow use of a variety of materials which offer both excellent wear properties and corrosion resistance.

Available sealing members of synthetic rubber, asbestos and Teflon complete the proper material selection.

Temperature

The temperature will in part determine the use of the sealing members. Synthetic rubbers are used to approximately 120°C, Teflon to 260°C, and asbestos to 340°C. Cooling the liquid in the seal chamber by water cooling jackets or cool liquid flushing often extends seal life and allows wider selection of materials.

Characteristics of Liquid

Abrasive liquids create excessive wear and short seal life. Double seals or clear liquid flushing from an external source allows the use of mechanical seals on these different liquids. On light hydrocarbons balanced seals are often used even though pressures are low to promote longer seal life.

Pump Model

Mechanical seal selection, installation and application varies with the pump model. All seals do not fit all pumps. Pump discharge pressures and temperatures are not necessarily the design conditions for the seal. Consult your pump sales engineer for selection of the proper seal for the service.

Installation Precautions

Protect all flexible members, as well as the seal faces. When passing the flexible members over sleeves, shafts, nuts, etc., cover threaded areas with paper or tape before sliding seal parts past, and check to be sure all burrs are removed and a small radius exists on the shaft sleeve ends. A light film of oil on sleeve surfaces will help greatly.

Check position of set collar, if used, to be sure that seal is confined to predetermined length when installed. When fitting single suction pumps with rubber bellow-type seals only, slide bellows on until it contacts its spring mechanism so that final step of mounting rotor in casing and tightening gland plate will push bellows to proper position. (Don't stop half way through this job). Bellows will adhere to shaft in position thus established and seal relies on bellows to prevent leakage between itself and the shaft or sleeve.

With double suction pumps, the seals have to be first put on the rotating element, and later on, this element is assembled in the casing. This can require too much elapsed time before the gland plates can be pulled into position. Therefore, it is well to make up a split collar to use instead of the gland plate, to position the seals with assembling on the rotating element. These collars are then removed and the gland plates pulled up when the element is assembled.

Mechanical seals are one item which should not be submitted to routine inspection maintenance. In operation the seal faces "wear in" with respect to each other, and if disturbed, it may not be possible to get them reseated. It is good practice to maintain a set of spare seal faces so that when pump maintenance causes the seals to be moved to get at other parts of the pump, these faces can be replaced.

Used faces can be replaced if returned to the seal supplier. The economy of this will depend on how special or large the seal may be. Roughly, if the seal is over 2 inches in size, relapping gets practical.

Also, if the seal is of the bellows type, spare bellows ought to be kept on hand.

A closing word about seals: insist that their lubrication be kept constant, clean and cool. Pressure must be slightly more than the differential across the seal. The seal liquid must have lubricating properties.

By keeping the sealant below 65°C seals will last longer even though manufacturers okay their application in up to 100°C water.

TROUBLESHOOTING

When discussing mechanical seal problems, the following analogy may be made: mechanical seals behave very much like bearings. Conditions that lead to bearing failure also lead to mechanical seal breakdown.

Misalignment

One of the main causes of mechanical seal failure is misalignment. This commonly occurs when seals are installed by personnel not trained in proper mechanical seal installation practices. If the seal is installed so that the mating faces are not parallel to each other and perpendicular to the shaft, the rotating face will try to square itself with the seat causing uneven wear on the mating faces.

To prevent this problem check to be sure that the seal is properly installed. The rotary unit must run parallel with the shaft and the stationary seat must be perpendicular to the shaft. Be sure the gland is drawn up evenly. This will eliminate the possibility of the improperly aligned stationary seat.

Lack of Lubrication

To operate properly, the mechanical seal mating faces must run on a liquid film. If there is no film, the seal will over heat and fail. Because this failure can occur in a matter of seconds, it is extremely important that the liquid film be present whenever the pump is in operation or about to be put into operation. Lack of lubrication at the seal faces is commonly experienced when handling light hydrocarbons near their vapour pressure. To ensure lubrication and dissipate heat, flushing must be provided at the seal faces.

Overheating

In some types of mechanical seal failures, overheating and lack of lubrication go hand-in-hand. This is especially true when the hard face heat checks. On examining a hard face that has heat checked you can see that the stress cracks are confined to the surface in direct contact with the carbon mating face. These surface cracks are caused when there is an intense heat build up on the surface of the hard metal, while the sub surface metal is not allowed to expand in a lineal direction because of the sub surface temperature, the surface buckles upward and cracks. The edges of the cracks, which are slightly raised above the face, begin to shave material from the carbon mating face.

Other symptoms of overheating are damaged elastomers. This condition is easily recognized because the elastomers will show cracking on the surface and harden.

To eliminate this type of failure, be sure the seal is operated in an environment having a lower temperature than the limits of any of its component parts.

Abrasive Damage

Abrasive damage is a common cause of seal failure. When minute abrasive particles work their way between the seal faces, they can groove the mirror finish of the faces or they can leave deposits from evaporation of the liquid. When these deposits build up on the shaft they will freeze the sealing elements to the shaft and eliminate all seal flexibility. This condition can be particularly acute when teflon sealing wedges are incorporated in the seal. Abrasive particles may become imbedded in the teflon, and can cause shaft wear directly under the sealing member. Abrasive liquids should be kept out of the stuffing box if at all possible.

Corrosion

Corrosion failure is easily recognized in most cases. Metal parts show signs of pitting, or the sections are totally corroded away. We will not attempt to go into a discussion of corrosion, but it should be remembered that the metal sections of mechanical seals are usually much smaller than the metal section of a pump. Therefore, they can withstand much less corrosion attack. It is a good rule not to down grade the metallurgy of the seal from that of the pump. When in doubt, select the more noble metal for the seals. (This is a general rule and does not hold true for all applications).

SUBJECT:

PREVENTIVE MAINTENANCE

TOPIC: 9

PUMP MAINTENANCE AND TROUBLESHOOTING

OBJECTIVES:

The trainee will be able to:

1. Recall the three basic categories of centrifugal and axial flow pumps.
2. Recall the installation and start-up procedures.
3. Recall the terminology used in describing the rotational axis of the pump.
4. Recall the installation procedures for each of the various types of pumps.
5. Recall the types of misalignment and how to check for misalignment.
6. Recall the steps to be taken in planning the disassembly of equipment.
7. Recall the principle of balancing a rotating element.

PUMP MAINTENANCE AND TROUBLESHOOTING

INTRODUCTION

The centrifugal and axial flow type pumps are probably one of the simplest of all mechanical assemblies having primarily only one moving part, i.e. the rotor.

Being so simple, they are the last to give trouble and, likewise the last to receive attention from a maintenance viewpoint. There are countless variations on the possible configurations, but all fall into 3 basic categories:

1) **Axial Flow** - where the direction of flow into the impeller and out of the impeller is parallel to the rotational centre of the impeller.
2) **Radial Flow** - where the medium being pumped has a change of direction from impeller inlet to outlet. The new direction finishes perpendicular to the rotation centre of the impeller.
3) **Mixed Flow** - where the flow is a combination of axial and centrifugal with the flow direction being somewhere between perpendicular and axial or parallel to the rotational direction.

INSTALLATION & START-UP

Where does maintenance begin? The ability of any piece of machinery to perform must begin with the attention given to the initial installation and pumps are as susceptible as any piece of machinery, to lack of expertise or attention in the process of installation. This is a key point and will have an important role to play in the eventual trouble free performance of the unit.

Let us then, as is said, begin at the beginning. The new unit arrives on site and now our responsibilities as operators and maintenance men begin.

It is not an exaggeration to say that many eventual pump problems begin right here. We must not only recognize what we are looking at, but how to approach it, understand it, treat it, install it, operate it and eventually and hopefully much later, maintain it and overhaul it.

Initial Observations and Checks on Arrival

1) Is the pump sealed on both its suction and discharge opening?

   Many problems are caused by foreign objects getting inside the unit before the installation even begins. Nuts and screws seem to be particularly attracted to the openings on pumps.

2) Is the pump well protected against corrosion, especially if storage before installation is anticipated?

   This is especially important on seal areas and glands.

3) If pump is packed and is not to be used immediately then leave the packing intact as much as possible.

4) Does the pump rotate freely?

   Having gone through the above and found everything satisfactory then leave the rest until installation is to be conducted.

Initial Installation: Pre-Checks

The procedures to be followed will vary according to the type of pump and the configuration of the particular installation.

The following general points must be observed:

1) Study the manual and any other literature which comes with the pump. Particular emphasis to be placed on installation instructions.

2) Thoroughly familiarize yourself with the pump you are about to install.

3) Do not presume it is exactly the same as number so-and so pump over at such and such station. Many pumps look alike, especially if from the same supplier, where as in fact, they may be quite different.

4) Check what equipment exactly will be needed for the installation. Pay particular attention to:

    a) What weights will you be required to lift and exactly how you will achieve each lift?

    b) What lifting equipment will be required, i.e. block & tackle, chain falls slings, eyebolts, etc.?

    c) What wrenches and other hand tools are needed? Are bolts metric or UNC, etc.?

    d) Do you know what torque figures are required for the various bolts?

    e) Will there be alignment requirements and if so, do you have the necessary shims or other material and gauges to achieve the correct alignment? Do you understand the alignment procedure?

    f) Are there any other special requirements including electrical or instrument work?

    g) Are there any specific safety requirements to be observed?

Installation of equipment needs carefully planning both from the standpoint of trouble free performance and also from the aspect of personal safety. Establish what is to be done and then plan carefully how it will be achieved.

BEFORE beginning any installation, be sure that all safety precautions have been taken:

a) Electrical lock out and tags have been attended to.

b) If working down in a pump pit or other "out of sight" location make sure that it is well known that you are down there and never go alone.

c) Suitable precautions relative to flooding have been taken if you are working on a system modification.

d) Any other hazards such as chemical or gas are identified and catered to.

e) Do not make presumptions, check and make sure it is safe to commence work.

A good idea is to make yourself a safety checklist during your planning stage and then check each item off before beginning work - **IT IS YOUR LIFE YOU ARE DEALING WITH - CHECK OFF THE ITEMS YOURSELF.**

Installation

It is impossible to cover all aspects of all types of installation. This topic will cover the more general points associated with the types of units you may encounter on your particular field.

The subjects to be covered are:

1) Horizontal close coupled
2) Horizontal with independent electric drive
3) Horizontal with independent internal combustion engine drive
4) Vertical turbine wet pit
5) Vertical dry pit
6) Vertical in line

Horizontal Close Couples (Fig 9-1)

One should start by being aware of the terminology. A reference to horizontal or vertical in pumps is related to the axis of rotation of the pumping member: i.e. **Vertical** means the rotational axis is vertical; and likewise, **Horizontal** means, as one may surmise, the rotational axis horizontal.

Close coupled refers to the prime mover being attached directly to the pump unit. These units may even have the impellers mounted directly onto an extended prime mover shaft, passed right into the pump case. Horizontal pumps, of the close coupled type are probably of the more simple type as far as installation is concerned.

Horizontal with independent drive both electric and internal combustion are more complicated and particular attention will be required relative to alignment.

Vertical Wet Pet would be where the pumping part of the unit is lowered into a pit which is flooded during normal operation. The discharge and the prime motive power unit will be above the water line and may even be on a separate floor level.

Vertical Dry Pit is a configuration where the pump is in a pit, normally below the water level, but the pit itself is not flooded. The water or whatever is being pumped will be piped into the pump suction.

Vertical In Line are as the name implies a unit which is installed directly into the pipe line.

Procedures and Considerations for Each Case

Horizontal close coupled pumps are probably the easiest to install, because:

i) They are pre-aligned during their manufacture.

ii) They are generally the smaller types of pumps.

1) If gaskets are to be used, then check that each gasket fits precisely the face onto which it is to be mounted.

2) Run over all bolts and nuts and check each is tight.

3) Prepare for lifting by installing eye bolts or other lifting devices where applicable. On eye bolts be particularly careful about thread engagement.

4) Check that bolts are capable of being fitted into all of the requisite holes.

5) Check connecting flanges are clean and free from damage.

6) Check and ensure that all piping systems have been cleaned and flushed through to remove any harmful contaminants or objects (i.e. nuts, bolts, 2"x4"s, etc.).

7) Lift pump onto its seat and check:

   a) Suction and discharge flanges can be connected without stress being applied to the pump case

   b) All bolt holes line up

   c) Mounting base holes line up

8) Determine if pump requires to be shimmed to align suction and discharge flanges. If so, install shims.

9) Insert all bolts finger tight into their appropriate holes.

10) Tighten flange bolts in sequence to ensure an even pull down of the mating flanges without distortion. Pay attention to the specified torque figures and do not exceed them.

11) Tighten base support bolts again to the required torque.

12) The electrical hook up to and instruments if applicable should be left to qualified personnel.

Horizontal Pumps with Independent Drive (Figure 9-2)

These pumps are somewhat more involved when it comes to installation.

The major difference is in the special consideration to be paid to pump and motor alignment which will be covered separately.

All of the steps previously outlined for the close coupled pumps will be applicable, but the following considerations will need to be taken into account:

1) If pump is mounted on a separate baseplate, then the baseplate should be installed before any attempt is made to align the unit (shims on both sides of each anchor bolt).

2) Baseplate should be installed in such a manner as to avoid imposing stresses which could result in twisting or deflection of the unit.

3) If baseplate is to be embedded in concrete then a non-shrink type must be used. Modern epoxy grouts are excellent for this purpose. Allow adequate time for setting of the concrete before the anchor bolts are pulled up tight.

4) Anchor bolts to be pulled up to specific torque before final alignment takes place.

Horizontal Pumps with Internal Combustion Engine Drive (Figure 9-3)

Again, all previous steps must be taken into account as far as the pump is concerned, but additional care must be exercised in relation to twisting of the baseplate. Internal combustion engines especially the diesel variety tend to be heavy and therefore the risk of twisting a baseplate is correspondingly greater.

Consideration must be given to engine controls, e.g. starting, running, cooling, fuel system, speed, exhaust, etc. All should be planned before commencing installation.

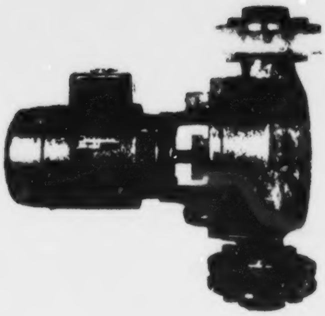


Figure 9-1 Horizontal Close Coupled Pump

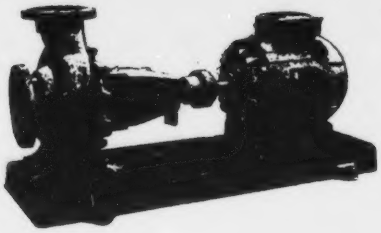


Figure 9-2 Horizontal Pump with Independent Drive

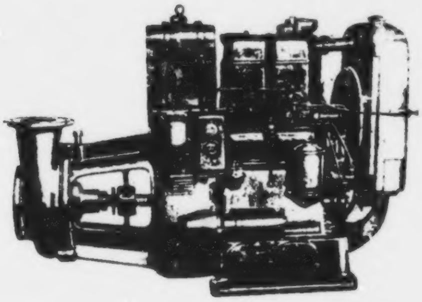


Figure 9-3 Horizontal Pump with Internal Combustible Engine Drive

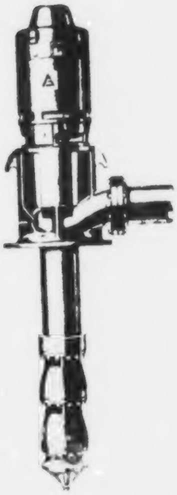


Figure 9-4 Vertical Turbine Wet Pit Pump

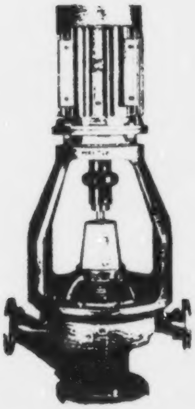


Figure 9-5 Vertical In Line Pump

Vertical Turbine Wet Pit Pumps will Require Special Consideration (Figure 9-4)

Where possible these pumps should be installed as a pre-assembled unit.

Problems can arise with the amount of crane lift necessary since this type of pump can be very long. Again, this should be considered at the planning stage.

It is important that such pumps be mounted in a true vertical position and from this point of view take great care, when setting the foundation plate.

A mistake often made is the radial alignment of the foundation plate which results in the pump facing in the wrong direction when the installation begins. This can be an extremely expensive mistake.

One method is to attach the foundation plate to the pump mounting and level pump and foundation plate as a unit with shims. This is by far the more desirable method and is less prone to mistakes being made. It is especially to be recommended if a seal is to be installed between pump base and function plate.

Before final pouring of concrete around the foundation plate, the alignment of the pump discharge flange should be checked. Avoid forcing discharge flange connections. This can cause severe stress to be transferred to the pump and result in premature flange or even discharge head failure.

Make sure when installing such pumps that objects are not dropped into the pump pit. Hard hats are especially prone to falling into pits and can cause some strange pump phenomena.

A further area of potential problems on this type of pump is lubrication. Most have water lubricated bearings and on many this water will be an independent supply. Adequate pressure and capacity should be checked as well as the quality. Many premature failures are caused by contaminants in the lubrication water supply.

Flushing the lubrication water lines is recommended before they are finally attached to the pump.

Previous comments on electrical hook up are equally applicable. Motor alignment will be covered separately.

Vertical Dry Pit will have the major change that they will normally have both discharge and suction lines to be attached and again care must be taken to avoid subjecting the pump to stresses caused by misalignment of the flanges.

Special safety should be considered, since in many cases the total pump will be below water level. Checks should be made to ensure that water pressure does not exist behind covers or under manholes which require to be disturbed during installation.

These pumps are generally shorter than their wet pit cousins and hence, crane height is normally less important, although still must be considered.

Vertical In Line (Figure 9-5) are probably the simplest of all to install and are also the most critical when it comes to the alignment of the suction and discharge piping. Where possible, piping should be flushed and cleaned before being connected to the pumps.

ALIGNMENT

This is probably the most critical phase of the installation.

Poor alignment can cause many problems and premature failure of rotating equipment.

Examples are:

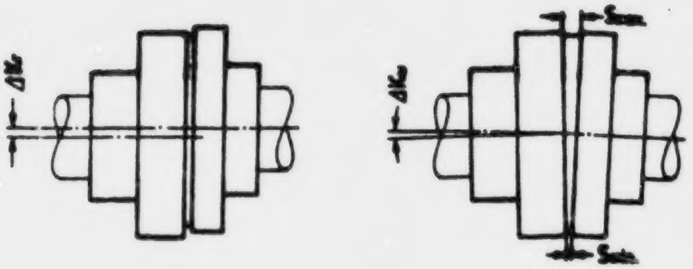
1) Vibration
2) Gland leakage
3) Bearing overheating
4) Bearing failure
5) Seal failure
6) Shaft breakage

There are only two basic types of alignment between rotat shafts and all others are a variation of these:

1) **Radial or sometimes referred to as Parallel or offset.** This refers to shafts which may be in line with reference to their rotational axis but offset to each other. Optimum alignment is achieved when the rotational axis are coincident with each other, i.e. zero offset. Refer to Figure 9.6

2) **Angular**

   In this case, reference is made to the angle existing between the rotational axis of each shaft. Refer to Figure 9 6.



Parallel Offset **Angular Misalignment**

Figure 9-6

RADIAL

Method 1 - Feeler or Taper Gauges

Refer to Figure 9-7 a & b. Feeler or taper gauges are inserted into the gap between the respective coupling halves and comparisons made between measurements. The measurements must be taken on two planes, i.e. horizontal and vertical to ensure all around correctness.

Shims, or in some cases, taper blocks are the normal methods used to correct misalignment of this kind.

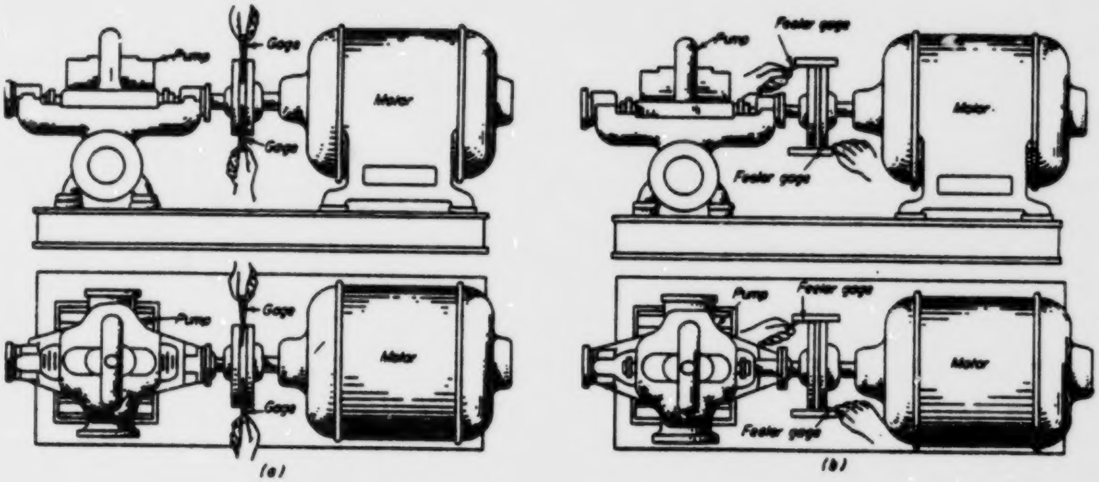


Figure 9-7 Checking Alignment with Taper and Feeler Gauges

Method 2 - Clock Gauges

On rigid couplings feeler gauges may not be sufficiently accurate to ensure acceptable alignment. In this case, clock gauges are the accepted alternative.

Clock gauges are also used in for example, the case where the coupling hubs may be just too far apart as to make feelers impractical. This would be so in the case of a spacer type coupling.

The clock gauges are set up as shown on Figure 9-8. The shafts are rotated together and the total reading is noted again in both vertical and horizontal planes. The actual misalignment in each case will be half of the total indicator reading.

The clock may be moved to the opposite flange and the process repeated as a check.

It should be noted that alignment using clock gauges is often made leaving one shaft stationary and rotating the other shaft. This is not as accurate since it does not take into account manufacturing machining errors on the coupling hubs and can result in expensive time consumption.

The above method of moving both shafts is particularly useful when working with very heavy pieces of equipment since it allows both shafts to be rotated independently but to a fixed measuring point on the coupling half.

On heavy equipment the process should be repeated in both directions of rotation and the average figure taken.

This will have the effect of balancing out clearness in bearings and bushes.

ANGULAR

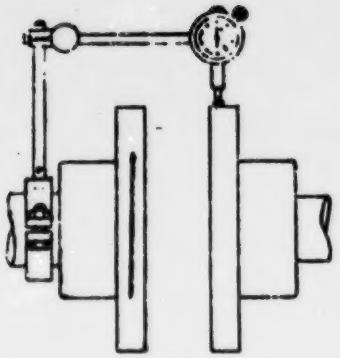
Method 1 - Feeler or taper gauges
Refer to Figure 9-7

As in the cause of the radial alignment gauges are inserted at four points on two axis to ensure both vertical and horizontal correctness.

Method 2 - Clock Gauges
Refer to Figure 9-8

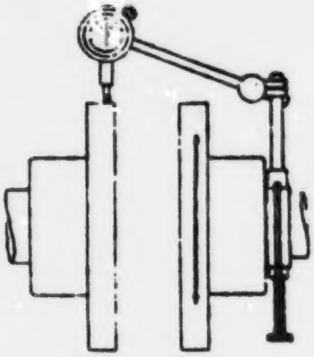
Again, the gauge is set up on one side and the shafts rotated together so that the same contact point is measured on both axis.

Rotation of both shafts together is often used but as previously mentioned with this method faulty machinery is not compensated for.

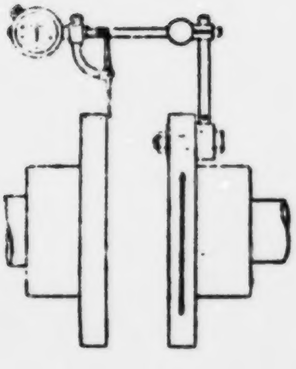


Radial Misalignment

1 a) Dial indicator mounted on driving shaft, rotate driving shaft to test concentricity of driven hub flange and shaft
Allowable +/- 0.573 mm
+/- 0.022 inches



1 b) Dial indicator mounted on driven shaft, rotate driven shaft to test concentricity of driving hub flange and shaft.
Allowable - +/- 0.573 mm
+/- 0.022 inches



Axial Misalignment

2) Dial indicator mounted on hub flange, rotate the hub to check regularity of gap between flanges.

3) Axial dimension 177.8 +/- 3.75 mm or 7" +/- 0.15 inches

Figure 9-8 Alignment Method by Using A Dial Indicator

General Notes on Alignment

1) Alignment should only be attempted when the unit is in its final site and has been grouted or otherwise anchored.
2) The required accuracy to be applied must be in accordance with the manufacturers specifications.
3) If, after aligning in one direction, it is necessary to correct the unit in the other direction, then the first alignment must be rechecked.
4) On new installed equipment alignment should be re-checked after a period of around six months. This ensures that changes due to settlement of foundations or thermal cycling are corrected.
5) Dowels should be used after final alignments to maintain the accuracy of the alignment process.
6) After completing final alignment, the couplings should be match marked to ensure correct assembly should further work become necessary.
7) For further information regarding alignment, please refer to Topic 12 in this manual.

DISMANTLING & REASSEMBLE

Removal

Major outage of important pieces of equipment must be planned just as carefully as the installation.

Account should be taken of:

1) Safety and hazard (chemical, electrical, flood etc.).
2) Equipment required (special tools, lifting, mount stools, etc.).
3) Planning sheet is extremely useful to determine the step by step procedure.
4) Before dismantling, unit should be examined to determine if match marks are visible. If not, marks should be added where appropriate.
5) The use of manuals should be made at every opportunity.
6) Take photographs before dismantling.

Dismantling

1) Careful attention to the order of dismantling must be paid to enable trouble free assembly.
2) Mark mating parts, if necessary.

3) Examine each individual item for telltale evidence of possible causes of failures (wear marks, discolouration, heat, etc.). This should be initially before cleaning. (Cleaning can sometimes remove or disguise valuable evidence).

4) Specific attention to be paid to:

   a) seals
   b) shafts
   c) bearing diameters
   d) bearings
   e) glands
   f) contamination of lubricant

5) It is rare to find a problem or failure that does not have some signs as to its cause. Train yourself to be observant.

6) Be careful not to accept what may appear to be obvious. Examine all parts with equal care.

Cleaning

1) Many cleaning fluids are available:

   a. Varsol
   b. Kerosene
   c. Trichloroethylene
   d. Carbon Tetrachloride

2) Care should be taken with many cleaning fluids regarding ventilation, fire hazard, and disposal of used fluids.

Final Examination

Should include:

1) Measurement of parts in wear (glands, wear rings, bearing diameters).
2) Indicator readings on rotating parts to determine if shafts are straight.
3) Tabulation is the best method of comparing dimensions. Make up sheet and document. This can be extremely useful for further overhaul in the future.
4) Make a material list of the parts required to complete the repair (include part numbers of components and full description, of pump including serial numbers).
5) Take account of any special process that may be required such as to mend casting cracks or welding bearing housings, etc.
6) Cavitation or erosion.
7) Finally, if pump requires to be stored until new parts are available, make suitable arrangements for rust protection and safe keeping of small parts.

REPAIR

Castings

1) Steel castings in general can be welded with very little difficulty, but care must be taken to avoid heat distortion of faces and bores.

2) Bearing housing bore diameters that have gone oversize can often be reduced by simple heating with oxy-acetylene equipment, (a simple metal shrinking process). Care has to be taken regarding possible distortion of faces.

3) Certain types of cast iron can be welded, but all types require special processes and should be left to specialists.

4) Oversized bores can be machined and bushed in many instances.

5) Many bronzes can be brazed to repair cavitation or erosion, but again on types such as aluminum bronze special processes may be required.

Shafts

1) Many shafts can be machined down to allow either sleeves to be fitted or alternatively to allow the fitting of undersize bushes. Particular care must be taken to avoid weakening the shaft, thereby causing the possibility of torsional failure.

2) Many methods are now available to deposit material on shafts enabling them to be re-machined to the original dimension.

These include:

- Plasma spray
- Arc welding
- Electrolysis
- Tungsten or carbide spray

Again all are highly specialized and must be taken to a suitable vendor.

3) Bent shafts can generally be straightened to an acceptable degree, but one should be aware of why the shaft has bent in the first place and remove the cause thereof.

Reassembly

Reassembly, in general, will be the reverse of the dismantling process, but with emphasis on cleanliness and protection from objects dropping into the pump.

Pay attention to:

1) Proper clearance of moving parts, especially bearing and wear rings.

Poor performance will result from overlarge wear ring clearance or, on open type impellers, large clearances at the impeller blades.

2) If gland type packing is used, it should not be pulled up tight until the pump is running. Be careful to provide all bearings, seals, packing and sleeves with adequate lubrication.

3) Proper alignment is critical and special attention should be paid as outlined under **ALIGNMENT**.

Balancing of the rotating element is a must, especially at the higher rotational speeds, i.e. above say 500 RPM. An under standing of what "Out of Balance" constitutes and how this causes problems is extremely useful to have. In basic terms, every rotating element has a "point of balance" or "centre of gravity" as it rotates (see Figure 9-9). The weight of the total mass acts through this point and it is theoretically possible to lift the whole element on a needle if applied directly under the centre of gravity (see Figure 9-10). Now, on any rotating element it is almost impossible to have the rotational axis and the centre of gravity coincident. When this happens, we introduce another element called "centrifugal force". This force causes a "skipping rope" effect on any rotating element and it exhibits itself as a vibration (see Figure 9-11).

In other words, when the centre of gravity is directly coincident with the rotational axis we have no centrifugal force, therefore, we are in balance. Conversely, when the centre of gravity is not coincident with the rotational axis, we have out of balance.

Vibration, in any form, causes many problems in rotating machinery, e.g.:

1) Premature bearing failure
2) Shaft fatigue and breakage
3) Oil leakage through seal failure
4) Loosening of fixing bolts and anchors

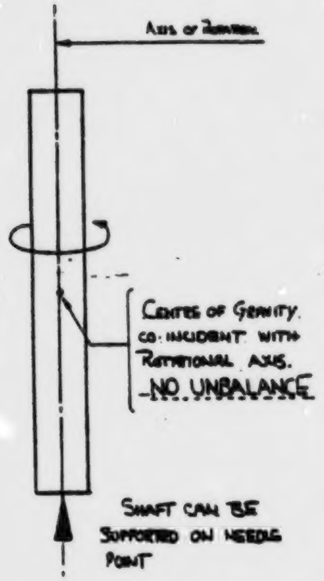


Figure 9-9 Balance Plate

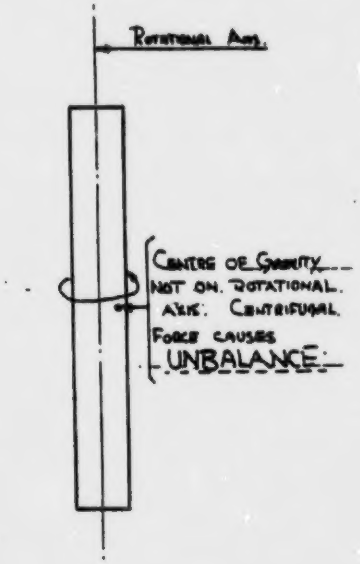


Figure 9-10 Balance Plate

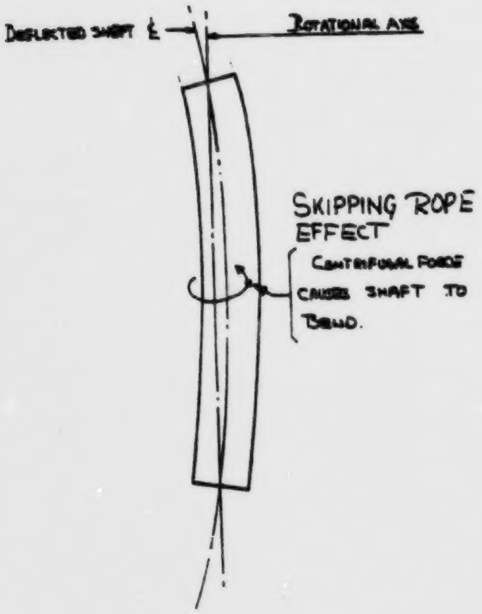


Figure 9 11 Balance Plate

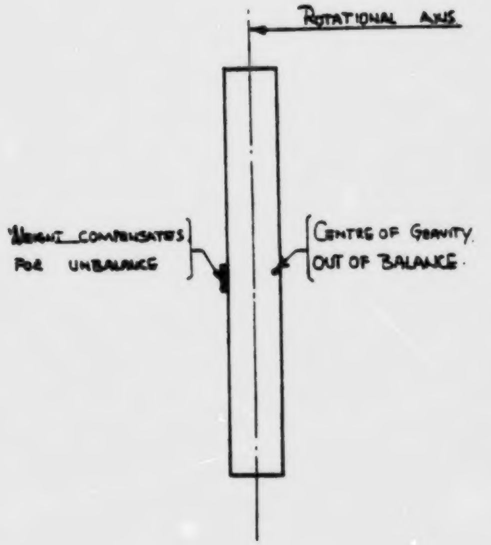


Figure 9-12 Balance Plate

Balancing a Rotating Element

It is very rare that the natural centre of gravity of a rotating mass can be modified to guarantee coincidence of centre of gravity and rotational axis. Another method must therefore be devised and this consists basically of compensation for out of balance load.

In other words, if we can determine the size and position of the out of balance load, it is reasonable to presume that the adding of a load of equal value in the opposite position will have the effect of cancelling out the out of balance lead, i.e. one will compensate for the other (see Figure 9-12). It is on this basic principle that balancing is conducted with weight being added or subtracted to give the requisite compensation. Within this frame work, we have two types of balancing procedures:

Static and Dynamic

1) Static Balancing is where the rotating element is suspended between two rollers having an approximation of "0" friction. The out of balance load will cause the mass to rotate under gravitation until the point of unbalance lies at the very bottom. Test weights are now added opposite to the lowest point until there is no rotation caused by gravity, i.e. the out of balance load has been compensated for resulting in **STATIC BALANCE.**

2) Dynamic Balancing is where the unit being balanced is rotated at a speed sufficient to cause the out of balance centrifugal force to exhibit itself. In this case, the rollers are fitted with instruments which measure the displacement, or amplitude as it is called, and the rotational speed of the mass we can determine the out of balance load. Further instruments on the machine determine the position of the out of balance load. It is best, when using this method, to balance at or close to the operational speed or slightly higher of the unit. It should be born in mind that weight can be removed to compensate for out of balance just as it can be added. The result is in any case compensation.

For best balancing results, a combination of both methods yields the best results.

PROBLEM SOLVING

A systematic approach is necessary to the definition of wha is the problem and hence how to solve it.

The following troubleshooting procedures may be helpful:

PROBLEMS IN CENTRIFUGAL PUMP OPERATION

No Liquid Is Delivered

Pump is not primed
Insufficient available NPSH
Suction line strainer is clogged
End of suction line is not in liquid
Impeller rotating in wrong direction
System total head is higher than pump
    total head at zero capacity

Pump Delivery Less Than Rated Capacity

Air leaks in suction line or pump seal
Suction line strainer partially clogged
    or of insufficient area
Insufficient available NPSH
System total head is higher than calculated
Partially clogged impeller
Liquid viscosity is higher than specified
Impeller rotating in wrong direction
Suction or discharge valves are partially closed
Impeller speed is too low
Impeller is installed in reverse direction

Pump Loses Prime While Operating

Liquid level has fallen below suction line intake
Air leak in pump or seal
Air leak in suction line
Liquid vaporizing in suction line

Pump Is Noisy

- Pump is cavitating
- Pump is misaligned
- Pump contains foreign material
- Pump shaft is bent
- Impeller is touching pump casing

Pump Is Vibrating

- Pump shaft bent
- Rotating element out of balance
- Bearings faulty

Pump Takes Too Much Power

- Impeller speed is too high
- Shaft packing is too tight
- Liquid is more viscous than specified
- Pump is misaligned
- Impeller is touching casing
- System total head is too low, causing too much pump delivery
- Liquid is denser than specified
- Impeller rotating in wrong direction
- Impeller installed in reverse direction

Noise & Vibration in Pumps is worthy of special mention.

Noise

1) Cavitation - recognizable as a deep slapping sound which can be noted to be travelling along in the discharge direction. It may even appear in a position away from the rotating part of the pump, e.g. in bends of pipework.

2) Pump is misaligned - is quite distinctly at rotational speed and will be most pronounced at housings close to the coupling. It has normally a low rumbling type of noise. Heavy vibration.

3) Contains foreign material - distinct scraping and highly pronounced grinding with higher pitch, not at a distinct speed (frequency).

4) Shaft bent - very similar to misalignment, but often accompanied by metal interference and heat. Frequency at rotational speed and constant.

5) Impeller touching casing - constant at rotational speed accompanied by heat and higher pitch sound, but not necessarily vibration.

Vibration

1) Pump shaft bent - frequency at rotational speed. Noise if present would be associated with mechanical interference between metals.

2) Out of balance - as for 1) but not normally accompanied by higher pitch noise. May disappear and reappear as pump is run up to speed and down to stop.

3) Bearings faulty - vibration signature will normally be of very high frequency range if rolling bearing. Heat will be present. Mechanical noise will almost certainly be present. Vibration will be pronounced at many frequencies.

VIBRATION IDENTIFICATION

CAUSE	FREQUENCY	REMARKS
Unbalance	1 x RPM	Most common cause of vibration.
Misalignment couplings or bearings and bent shaft	1 x RPM usual 2 & 3 x RPM sometimes	Best found by appearance of large axial vibration. Use dial indicators or other method for positive diagnosis. If sleeve bearing machine and no coupling misalignment balance the rotor.
Bad bearings anti-friction type	Very high several times RPM	Bearing responsible most likely the one nearest point of largest high-frequency vibration.
Eccentric journals	1 x RPM	If on gears largest vibration in line with gear centers. If on motor or generator vibration disappears when power is turned off. If on pump or blower attempt to balance.
Bad gears or gear noise	Very high gear teeth times RPM	
Mechanical looseness	2 x RPM	Usually accompanied by unbalance and/or misalignment.
Bad drive belts	1, 2, 3 & 4 x RPM of belts	Strob light best tool to freeze faulty belt.
Electrical	1 x RPM or 1 or 2 x synchronous frequency.	If vibration amplitude drops off instantly when power is turned off cause is electrical.
Aerodynamic hydraulic forces	1 x RPM or number of blades on fan or impeller x RPM	Rare as a cause of trouble except in cases of resonance.
Reciprocating forces	1, 2 & higher orders x RPM	Inherent in reciprocating machines can only be reduced by design changes or isolation.

SUBJECT:

PREVENTIVE MAINTENANCE

TOPIC: 10

MAINTENANCE OF STAND-BY POWER UNITS

OBJECTIVES:

The student will be able to:

1. Describe the purpose of the automatic transfer switch.
2. Recall the check list used for routine maintenance of an emergency power system.
3. Recall the maintenance guidelines regarding fuel and engine oil filters.
4. Recall the common causes of turbocharger failure.
5. Recall the minimum operating conditions for the diesel engine for simulation of power failure.

MAINTENANCE OF STANDBY POWER UNITS

Preventive maintenance of a standby power unit is a very essential matter if one is to rely on the unit when an emergency arises. The intention of this topic is an attempt to acquaint the operator with a basic understanding of generator sets, with routine preventive maintenance procedures that should be conducted as well as some troubleshooting points.

TRANSFER SWITCH

The automatic transfer switch may be referred to as the "heart" of the system. When there is a power failure, it is up to the automatic transfer switch to sense the failure, initiate the starting of the emergency generation, and put it on line at the proper time to supply power to the loads. If the transfer switch does not function properly, the loads could be left without power no matter how elaborate the rest of the system may be. A good preventive maintenance program will ensure the reliability of the transfer switch and related elements.

Installation Precautions

Preventive maintenance begins with the installer when the automatic transfer switch arrives at the job site. Check for shipping damage. The installer should closely follow the manufacturer's instructions. After the equipment is installed, it should be thoroughly cleaned and vacuumed. Before energizing the circuits, the switch should be manually operated in accordance with the manufacturer's instructions.

Next, the phase rotation between the normal source and the emergency source should be checked. Now the switch can be energized and given a functional test in accordance with the manufacturer's instructions.

Testing

Regular testing of an emergency power system will uncover potential problems which could cause a malfunction during an actual power failure. Furthermore, regular testing gives maintenance personnel an opportunity to become familiar with the equipment and observe the sequence of operation. By being familiar with the equipment, personnel on duty during actual power failure can respond more quickly if a malfunction occurs.

An automatic emergency power system should be tested under conditions simulating a normal source failure. Experience shows that the rate of discovery of potential problems when a system is tested automatically with the emergency generating equipment under load for at least one-half hour, is almost twice the rate when a system is tested by manually starting the generating equipment and letting it run unloaded.

A test switch is recommended on the transfer switch to simulate a normal power failure. Some manufacturers supply this as standard equipment. The advantage of a test switch is that it does not cause a loss of power to connected loads as is caused if the main disconnect is opened to simulate a failure. A test switch opens one of the sensing circuits and the controls react as if there were a normal power failure. In the meantime, the loads stay connected to the normal source. Power is interrupted to the loads only when the load is being transferred from the normal to the emergency source and visa versa. This interruption is only a few cycles.

On the other hand, if the main disconnect is opened to simulate loss of normal power, then the power to the load could be interrupted for up to ten seconds. The interruption could be much longer if a malfunction occurs during start-up of the emergency generator. When a test switch is used, fewer maintenance personnel are needed because the potential of experiencing an extended power outage during the test is significantly reduced.

A minimum check list for an emergency power system should include at least the following items:

1. Loose control wires and cables
2. The time from when the test switch is activated until the engine starts to crank

3. The time required for the engine to achieve nominal voltage and frequency after the test switch is activated
4. The time that the transfer switch remains on the emergency generator after the test switch is released
5. The time that the engine generator set runs unloaded after the load circuits have been transferred back to the normal source of power
6. Proper shutdown of the engine generator set and engine auxiliary devices

Inspection

In addition to putting the system through test procedures, visual inspection is also necessary to guard against system malfunction.

Before starting a scheduled operational test, the equipment should be checked for loose control wires and any discolouration of power connections caused by excessive heat due to loose connections. Also inspect for excessive dust and dirt particles and "craters" left in the dust by excessive moisture. With proper care, this type of inspection can usually be done without de-energizing the equipment.

If the transfer switch is subject to a fault current resulting from a short in the distribution system, the switch should be checked immediately to determine if it was damaged. Such damage can be a potential source of fire, jeopardize the safety of personnel, or adversely affect the operation of the switch. To check the switch for such damage, first de-energize the switch by opening both the normal and the emergency breakers, and then look for any of the following:

1. Damage to insulating parts
2. Evidence of arcing between live parts and to ground
3. Welded or significantly eroded main and/or arcing contacts

While the switch is de-energized a dielectric test should also be conducted to determine if the switch will withstand twice the rated voltage plus 1000 volts rms. The length of downtime for this inspection is kept to a minimum if the design of the transfer switch allows inspection of the live parts without major disassembly or removal of switch sub-assemblies such as a portion of the drive mechanism, etc.

Detailed Inspection Suggestions

An in-depth inspection should be performed at least annually. This must be done with the equipment de-energized.

Main Operating Mechanism

The main operating mechanism of an automatic transfer switch should be rugged and simple and lubricated for life. The parts that are lubricated should be checked at this time. If, due to environmental conditions, the mechanism requires lubrication, the manufacturer's recommendations should be followed.

Contacts

C.S.A. standard C22.2 No. 178 requires that an automatic transfer switch be capable of from 2000 to 6000 cycles of operation under load, depending on amperage size, without parts replacement. However, the main current carrying contacts should be checked during this inspection. To do this easily, the pole covers and arc chutes should be able to be removed without any disassembly of the operating mechanism.

Expensive time is consumed if the operating mechanism must be lubricated and adjusted after it is reassembled. The best rule to follow is to choose equipment that does not require major disassembly for inspection and preventive maintenance. Contacts on transfer switches that employ integral overcurrent protection cannot be conveniently inspected.

Desirable Features

To make inspection easy and safe, it is desirable to have the following additional features in the transfer switch:

1. Transfer switches rated 100 amperes or larger are usually mechanically held in both the normal and emergency positions. In such switches the main contact pressure depends on the mechanical locking mechanism of the main operator. Automatic operation of these switches in either direction requires electrical control power. Thus, the main poles of the transfer switch should be the type that are held closed mechanically and energized. In this type of design, the load remains energized while the sensing and control circuitry is de-energized for inspection.
2. Sensing and control circuitry on a panel separate from the main transfer switch panel.
3. Compact and light-weight sensing and control panels to allow for mounting on an enclosure door that swings out and away from the energized main poles of the transfer switch.
4. A harness containing a disconnect means to allow the sensing and control panels to be completely de-energized.

With the sensing and control panels de-energized, the control relay contacts, time delays, and pick-up and drop-out values of the sensing circuits can be checked.

With the harness disconnected, the transfer switch is incapable of automatically responding to a power outage. Therefore, emergency procedures must be written and personnel must be ready to respond if an emergency should occur. Being prepared, personnel can save precious time in manually transferring the loads to the acceptable emergency source.

Preventive Maintenance and Repair

Dirt, dust and moisture cause most malfunctions in automatic transfer switches. Eliminating these is the best preventive maintenance that can be performed. Dust should be removed by a vacuum means during the annual inspection.

Do not use compressed air because the flying dust and other particles can settle on uncovered control relay contacts and armature surfaces, and adhere to lubricated surfaces. If condensation moisture is found to be a problem, a small space heater can be used.

Replacement Parts

There are no consumable spare parts in an emergency power transfer system. However, replacement parts should be kept on hand in case of malfunction occurs. The quantity of replacement parts will be determined by the following factors:

1. Manufacturer's recommendations
2. Replacement parts common to all installed equipment. If a majority of the manufacturer's recommended replacement parts are common to all transfer switches on the premises regardless of type, system voltage or amperage size, then a minimum quantity of replacement parts are required. For example, if all control relays are identical and are powered from the secondary of a control power transformer, the same coil kit can be used on all control relays.
3. Availability of replacement parts from a local source such as a stocking distributor. However, if loads are critical, parts should be purchased and kept on hand.

Each replacement kit should be opened and the contents checked for the correct parts when they are received. The installation instructions should be studied so that maintenance personnel know how to change the parts. No special tools should be required to install the parts. An inventory list of the spare parts kits should be kept.

FUEL OIL

Diesel fuel oil should not exceed 0.2% sulphur. A product of combustion is water and when combined with Sulphur Dioxide, Sulphuric Acid is formed which causes damage to bearings, pistons, etc.

Diesel fuel should have a good lubricating quality. Great care must be taken in the choice of a fuel additive, if one is to be used at all. Combustion deposits can be formed, thereby shortening the engine life.

Fuel Oil Filters

The manufacturer recommends fuel filter changes after 7,000 gal. of fuel being used or 500 hours of operation. The maximum time in use should not exceed 12 months. Clean the water trap at the same time.

Engine Oil and Filter

The maximum engine running time between engine oil and filter changes should not exceed one year.

Before Starting the Engine

Before simulating a power failure the engine coolant and lube oil should be checked. Some engine dip sticks are marked engine "Stopped" and engine "Running". Engines with a

Woodward governor have a separate reservoir and it has its own dip stick. Use the same oil as used in the engine.

Check the battery electrolyte level. Do a visual check of the engine.

After the engine is started and is running, check the engine coolant temperature, the oil pressure, oil temperature and the engine RPM. Check the fuel oil pressure (50-75 psi). Check that the battery is being charged. Check the Hertz - it should be 60 under load, 62 under no load. Do a visual check of the engine.

Air in Fuel System

If for some reason air has entered the fuel system it will have to be removed prior to operation. Start to bleed the fuel tank filter and water trap. Bleed until all the air is removed and there is a full flow of fuel.

To bleed the engine fuel filters, use the priming pump on the fuel injection pump or on the engine. Bleed the primary and secondary filters until there is a full flow of fuel, free of air. Crack open the bleed valves on the injection pump and bleed until free of air.

Start the engine to be sure there is not any air left in the system which will cause it to stop. If the engine runs for a short period and stops, check for leaks, check all fuel line fittings. Air can be drawn into the system through any loose fitting or filter gasket thereby preventing the flow of fuel.

If the fuel system is satisfactory but the engine does not start, crack the high pressure fuel line fitting and the injectors. If you are still not getting any fuel it is likely there is a problem with the injection pump or the primary lift pump.

TURBOCHARGERS

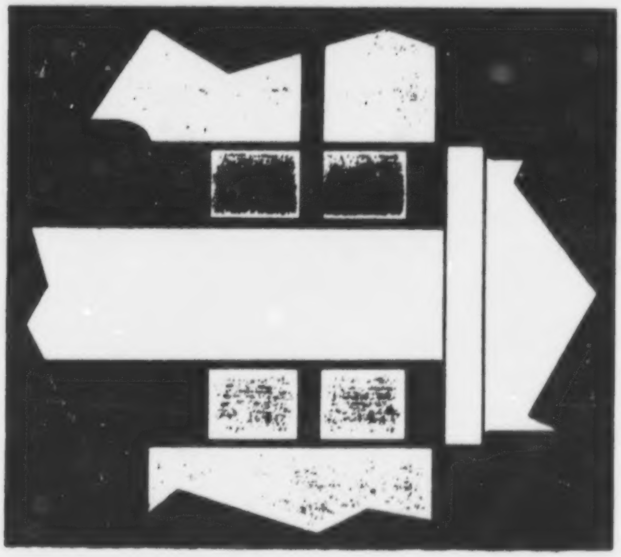
The turbocharger is driven by the engine exhaust gases. The efficiency of the turbocharger is increased as the load increases because of the expansion of the gas. The exhaust side is called the "turbine". The intake side is called the "compressor".

The advantages of the turbocharger are increased horsepower, fuel is more completely burned, there is less exhaust noise and generally a cooler exhaust stack.

Since turbochargers are lubricated by the engine oil, if the engine is using excessive oil, check the turbine end and compressor end for oil leaks. The seals could be leaking.

Turbocharger Failure: Common Causes

Many failures are attributed to "lack of lubricant". Lubricant flow to the journal and thrust bearings must be adequate to cool the bearings, lubricate and clean the surfaces. The shaft and journal bearings "float" in a film of oil, and there must be sufficient oil or shaft motion can result.



Shaft and journal bearings "float" in a film or oil.

Lack of lubricant directly causes shaft motion and damage to both journal and thrust bearings. Once the shaft is unstable, and rotating "off centre", there can be secondary damage: wheel contact with housings; bent or broken shaft; destruction of seals and thrust plates.

Determine the Cause of Lack of Lubricant

It is not sufficient to merely determine the lack of lubricant was responsible for a turbocharger failure. The problem must be traced to its source and corrected.

First determine the pattern of oil flow inside the turbocharger (Figures X and Y). On type X, the journal bearings receive oil directly through the central oil cavity of the turbocharger. The thrust washer receives oil by an individual passage. A lack of lubricant failure of the journal bearings would probably mean that the entire turbocharger was starved of oil. A lack of lubricant failure of the thrust bearing alone, could indicate a blocked passage.

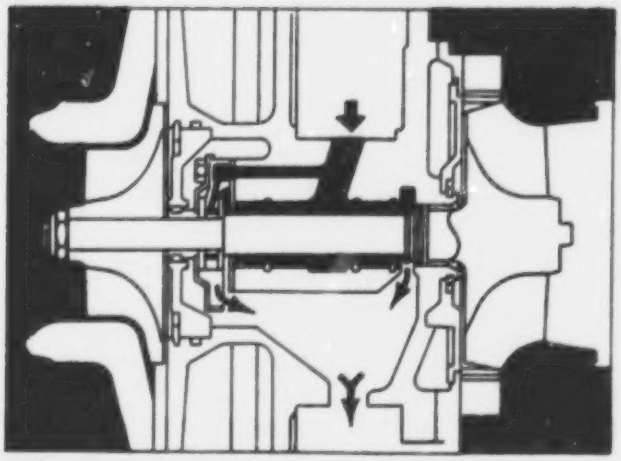


Figure X

On the type Y turbochargers, all bearings are lubricated via the smaller oil passages. Oil flows through the bearings too, via lubrication holes. With this lubrication design, a blocked passage or damaged oil holes could affect any one bearing without affecting others. On this type of turbocharger, examining all the oil passages is very important.

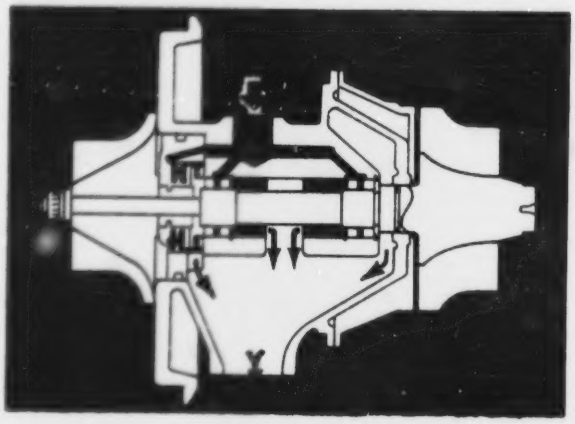


Figure Y

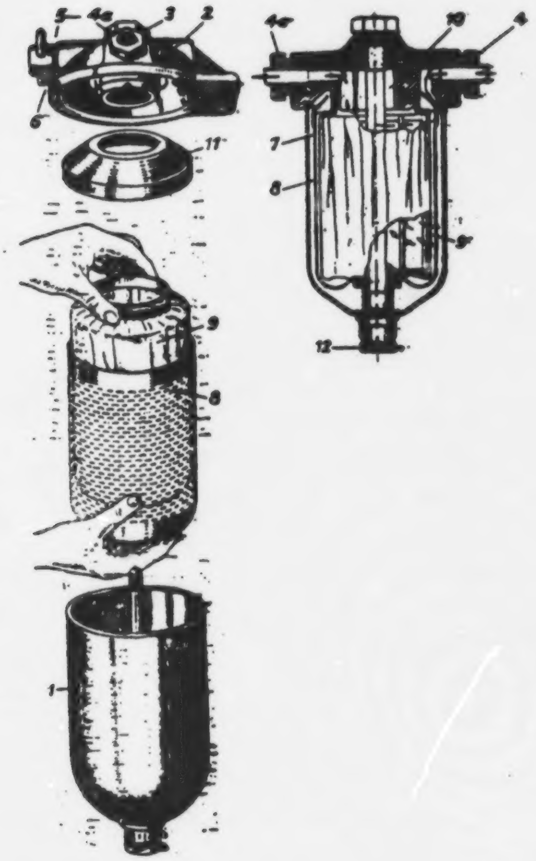
Continue your search into the engine for the cause of lubricant shortage. Look for:

- Malfunctioning turbocharger lubrication valve, if the engine is so equipped
- Low engine oil pressure
- Blocked oil supply lines
- Improper weight of oil
- Find out if the turbocharger was properly prelubricated when installed.

Fuel Filter (Bosch Type)

For proper operation, the fuel entering the combustion chamber must be clean or else unburned particles will remain. Dirt will also cause clogging of the spray valves and prevent the full quantity of fuel from entering the combustion chamber. Fuel oil is often screened four or five times in its path from the tank to the tip of the spray valve.

The illustration shows a form of fuel filter. Fuel enters by gravity or pressure from the fuel tank through inlet 4; it flows down and surrounds the screen 8, through which the clean oil passes - leaving the impurities and heavier particles at the bottom; it then passes through a cloth filter 9, and out through outlet 4a.



It is well to install a filter close to the injection pump in order that it may catch scale and other deposits in the piping. A cut off valve between the tank and filter permits opening and cleaning the filter. A bleeder screw 5, is turned for an instant to allow trapped air to escape; the heavy oil and impurities are removed by opening the drain plug 12.

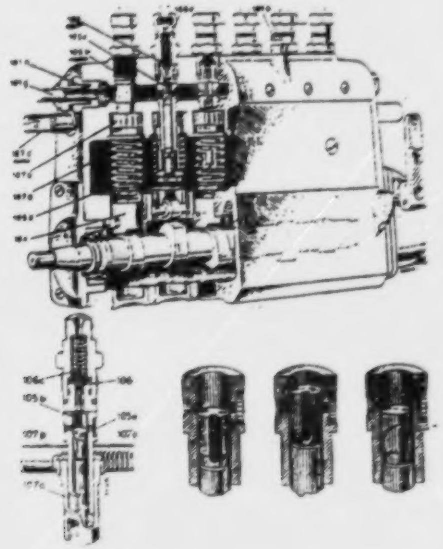
To clean the filter the engine is stopped and the fuel supply cut off. The nut 3, is loosened and the outer shell 1, drawn down, then the cover 11 is removed and the metal screen and fabric filter taken out; these are thoroughly brushed with a not too hard brush and washed with gasoline or kerosene. The entire is then reassembled. The bleeder screw is opened to permit the fuel to force out the air.

Fuel Control

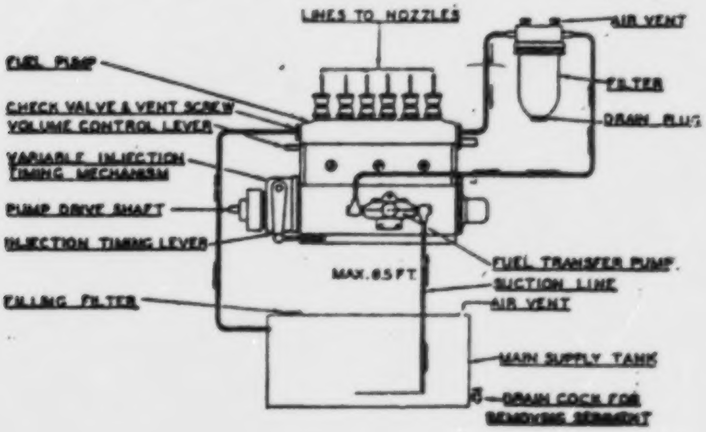
The control of the quantity of fuel delivered by a pump varies in accordance with the design of the pump and valves - the pump plunger may be varied in stroke or the fuel by-passed.

Fuel enters the pump barrel through the inlet or opening above the plunger as shown at 105b (Bosch pump). The plunger, in rising, forces this fuel up through the valve 106; the plunger, however, can be rotated at 107 d, so that the helical cut permits some of the fuel to flow down the groove and out through the port at the side, lessening the quantity ejected. The control can be readily seen by the varied positions of the plunger in relation to the outlet port.

(Figures - 2)



INDIVIDUAL PUMP FUEL INJECTION SYSTEM



In the individual pump injection system, fuel is supplied to the cylinders by individual pumps operated from cams on a camshaft or on an auxiliary drive and are independent from each other in their operation.

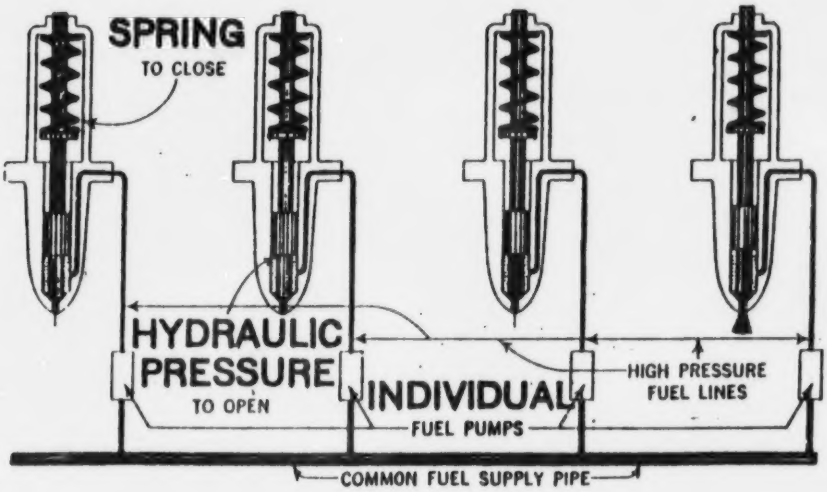
Fuel is drawn from the main tank by a fuel supply or transfer pump and passes through a filter into a day or operating tank, or directly to the pump supply pipe. From there it is taken up by the pump plungers and injected into the cylinders in quantity and time pre-arranged. Such an arrangement is shown above - the particular piping layout being that of the OTTO ENGINE WORKS - SUPERIOR DIESEL ENGINES.

The pump plunger strokes are usually of the constant stroke type and adjustment made by by-passing some of the oil into the supply tank - the method of by-passing being different with different pumps. Individual pumps can be taken out of service without affecting the remainder, and pressures in the system are low except when a plunger is injecting fuel into the cylinder and then only is there a pressure between pump plunger and cylinder.

Filtering the fuel oil is a very important function in engine operation. Pumps are finely adjusted, have very little play between parts and would soon clog if dirt and impurities were to enter the barrels. Pumps must send a definite amount of fuel, at a definite pressure, into the engine to maintain it at proper speed and power and for uniform cylinder pressure.

FUEL PUMPS

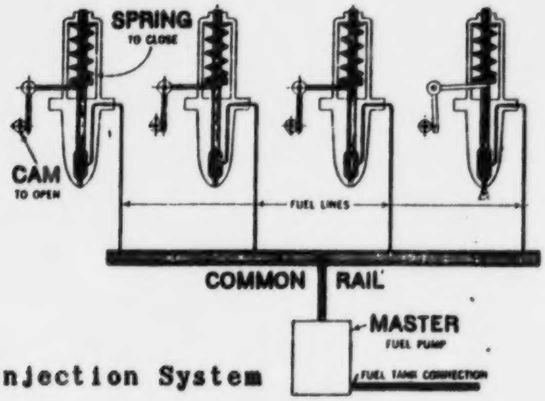
Hydraulic Spray Valves



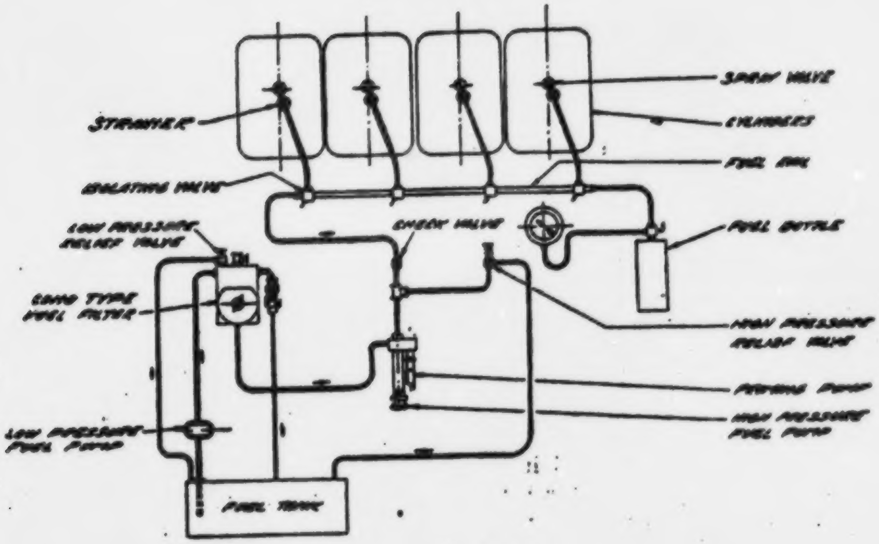
Elementary individual pump fuel injection system. In his system a separate pump is used for each cylinder. The pumps work at pressures of from 1,000 to 10,000 lbs. per sq. in. Metering of fuel and timing of injection is done within the pump unit, the pump of each cylinder discharging through its connecting line and injection valve at the proper time. The hydraulic valves are opened by oil pressure and closed by a spring.

Elementary master pump "common rail" fuel injection system. The basic idea is the use of one high pump to supply all cylinders. The mechanical spray valves are opened by cam gear and closed by a spring.

Mechanical Spray Valves



Common Rail Fuel Injection System

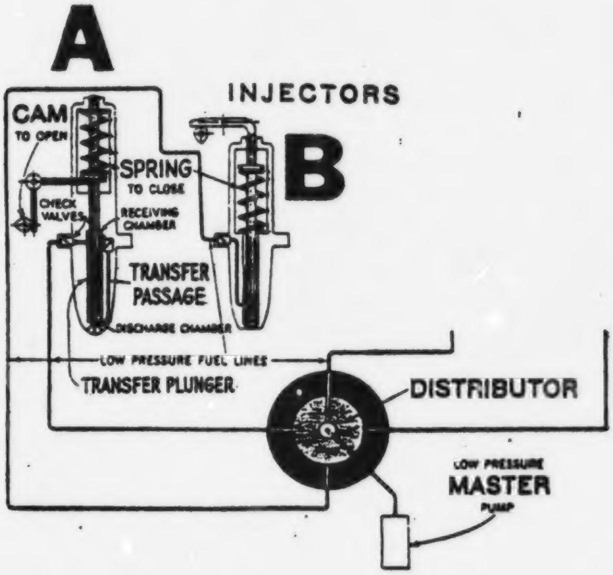


In the common rail system, fuel is supplied to a fuel pipe or manifold common to all cylinders. A high pressure pump maintains a constant pressure in this manifold or common rail, as it is called, from which taps are taken off to feed the combustion chambers of the various cylinders. Fuel is drawn from a tank, through a filter to an operating or day tank by a low pressure suction pump; from there the fuel travels to the suction side of the high pressure pump from where it is raised to the required operating pressure - 3,000 lbs. to 5,000 lbs. per sq. in. or more.

A high pressure relief valve, adjustable to various pressures, maintains the pressure constant in the fuel rail and spray nozzles. If the pressure increases beyond a set point, the valve opens and permits some of the fuel to flow back, or by pass, into the tank. This system is also called the "constant pressure system". A low pressure relief valve by-passes the fuel on the low pressure or transfer pump side. Check valves prevent the return of the fuel by cylinder or back pressure. Cut off valves cut off the feed to a cylinder if it becomes necessary to do so for repairs or replacement of parts or nozzles. An accumulator or fuel bottle, serves as a cushion to smooth out the pulsations produced by the pump. Pressure gauges indicate the pressures in the line.

The diagram shows the common rail arrangement used by ATLAS IMPERIAL DIESEL ENGINE CO. The fuel nozzles are opened mechanically and permit the fuel under pressure to pass through.

Injectors



Distributor system with unit injectors illustrating A, Graham mechano, acto transfer injector and B, reversed cone nozzle injector.

These injectors are opened by a cam and closed by a spring.

In the operation of the Graham injector as the valve closes the master Pump forces (via distributor) a metered charge of fuel into the receiving chamber at low pressure. During the opening of the valve, this charge is transferred to the discharge chamber where it is pre-heated and discharged through the multi-nozzle passages during the closing of the valve. This injector requires no valve adjustment and no high pressure fuel line.

Note in injector B that the reversed cone valve moves downward to open.

NOTE:

According to Research Laboratories Section of General Motors, the pressure in injector pumps may reach 20,000 lbs. per sq. in. Under the usual conditions of operation the pressure is 15,000 lbs. per sq. in. The pump forces the fuel through small holes in the spray tip at 13 miles a minute - 780 miles an hour. The high velocity breaks the oil up into a fine penetrating fog.

ENGINE OPERATION

The diesel engine should be operated once a week preferably at full load, but at least at a minimum of 30% of rated power on the engine.

The normal source or main breaker disconnect should be opened to allow the standby to start up in normal power failure condition. To shut down the engine, close the normal source disconnect. This will assure that all start up and shut down components are functioning properly.

The engine should be run under load for one hour to allow the coolant and lube oil to obtain normal temperatures.

SUBJECT: **TOPIC: 11**

PREVENTIVE MAINTENANCE **MOTORS**

OBJECTIVES:

Trainee will be able to:

1. Discuss the factors to be considered when selecting an electric motor as the driving power for a pump.

2. Recall the names and characteristics of various motor enclosures available for different pumping environments.

3. Recall the names and characteristics of two types of AC motors commonly used with pumps.

MOTORS

FACTORS IN MOTOR SELECTION

Pumps are usually driven by electric motors, but may also be driven by internal combustion engines using diesel oil, gasoline, or a liquified gas as the fuel. In some cases, steam turbines, water turbines, or gas turbines may also be used as sources of power to drive pumps. However, the most common driver, at the present time, is the electric motor and it is mainly towards this device that this topic is directed. Electricity is usually the most economical and convenient source of energy. Factors to be considered when selecting an electric motor are:

1. Driving power required by the pump
2. Speed of rotation of the pump
3. Environment in which the pump must be operated
4. Motor types

Driving Power

The motor must be sized so that the total power consumed by a pumping system is satisfied. The total power is the sum of the power required to:

1. Lift the liquid from its present level to the new level

2. Overcome friction in the suction line, discharge line, pump intervals, bearings, shaft seals and packing glands

3. Compensate for the power losses in the drive train between the motor and pump

Normally, pump drives should be overpowered by about 10%.

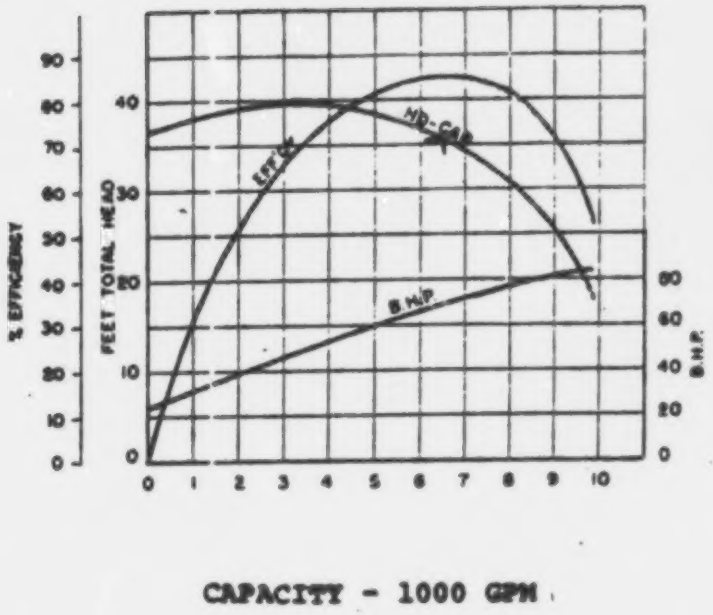


Figure 11-1 Characteristic Pump Curve

In the case of centrifugal pumps, the pump curve should be consulted to ensure that the motor size selected will not cause a motor overload due to a pump "run-out" condition caused when the total head imposed in the pump is decreased. This is illustrated in Figure 11-1. The maximum efficiency determined by the ratio of head capacity to brake horsepower is 70% when the head is 36 feet, the capacity is 6500 gpm, and the motor required is 70 HP. However, if the pump is used for anything other than normal duty during which the total head of the system decreases to 30 ft., the pump capacity will increase to about 8,000 gpm and the HP requirement will increase to about 80 HP. Thus, this possibility can be avoided by installing an 80 HP motor at the time of the original installation.

Speed of Rotation of the Pump

Generally, centrifugal pumps have specific speeds of less then 3,000 rpm. Thus motor speeds of 3,580 rpm provide the most efficient unit because the specific speed varies directly with the motor speed, and efficiencies increase with increasing specific speed (up to approximately 3,000 rpm specific speed). In addition, the higher the speed of the pump, the smaller and more compact the unit becomes. Accordingly, if efficiency and cost per gallon per minute pumped were the only basis of consideration, the above speeds would be common. However, other factors must be considered such as noise, vibration and suction conditions. When these factors are considered, lower speeds of rotation are indicated rather than the expenditure which would otherwise be required to correct the problems created by higher more efficient pump speeds.

Thus, for smaller pumps, the more common speed to use is a nominal 1,800 rpm (4 pole motor). For sewage and sludge pumps especially, a nominal 1,200 rpm (6 pole motor) speed is usually used. For large pumps, both for water and wastewater pumping, a nominal 900 rpm (8 pole motor speed) is preferred.

Environment

The ideal environment for a pump motor is in clean, dry air - free from dust and dirt, with a low humidity and at an ambient temperature of about 20°C. All of these conditions, are not normally present in one location in water and wastewater treatment plants and pumping stations. However, some of the conditions can be provided at moderate cost at the time the plants and pumping stations are designed. When the environment is less than ideal or when flammable gases may be present, the type of motor enclosure most suited to the conditions prevailing, must be given careful consideration. The specialized types of motor enclosures are more expensive but in the long run can save costly maintenance repairs.

Many motor enclosures are available which will permit operation of an electric motor in a wide variety of environments. The specific application and the nature of any contaminants present will dictate the best enclosure for the job.

Moisture problems require special consideration. Suitable guards or enclosures must be provided to protect exposed current-carrying parts of motors and the insulation of motor leads where dripping or spraying water, oil, or other injurious liquids may occur, unless the motor is specially designed for the existing conditions.

The following are the names and characteristics of motor enclosures available:

1. Open Enclosures
    a) General Purpose. Openings to permit passage of external cooling air over and around the windings of the machine for ventilation.
    b) Drip-Proof. Ventilation openings are so constructed that successful operation is not interfered with when drops of liquid or solid particles strike or enter the enclosure at any angle from 0 to 15 degrees downward from the vertical.

c) **Splash-Proof.** Ventilation openings are so constructed that successful operation is not interfered with when drops of liquid or solid particles strike or enter the enclosure at any angle not greater than 100 degrees downward from the vertical.

d) **Guarded.** Openings giving direct access to live or rotating parts (except smooth shafts) are limited as to size by the design of the structural parts or by screens, grills, expanded metal etc., to prevent accidental contact with such parts.

e) **Semi-Guard.** Part of the ventilation openings, usually in top half, are guarded as in a "guarded machine", but others are left open.

f) **Drip-Proof Fully Guarded.** A drip-proof machine with ventilation openings as in a "guarded machine".

g) **Externally Ventilated.** A machine which is ventilated by means of a separate motor-driven blower mounted on machine enclosure. Mechanical protection may be as defined above.

h) **Pipe Ventilated.** Openings for admission of ventilating air are so arranged that inlet ducts or pipes can be connected to them.

i) **Weather-Protected - Type I.**
Ventilation passages are so designed as to minimize the entrance of rain, snow and airborne particles to the electrical parts.

j) **Weather-Protected - Type II.**
Ventilation passages at intake and discharge are so arranged that high-velocity air and airborne particles blown into the machine by storms or high winds can be discharged without entering the internal ventilation passages leading directly to the electrical parts.

2. Encapsulated/Sealed Enclosures

a) **Encapsulated Windings.** An alternating-current squirrel-cage machine having random windings filled with an insulating resin which also forms a protective coating.

b) **Sealed Windings.** An alternating-current squirrel-cage machine making use of form-wound coils and having an insulation system which, through the use of materials, process, or a combination of materials and processes, results in a sealing of the windings and connections against contaminants.

3. **Totally Enclosed**

a) **Non-ventilated.** Enclosure prevents free exchange of air between inside and outside of case, but not airtight; not equipped for cooling by external means.

b) **Fan-Cooled.** Equipped for exterior cooling by means of a fan or fans, integral with machine, but external to the enclosing parts.

c) **Fan-Cooled Guarded.** All openings, giving direct access to fan, are limited in size by design of structural parts or by screens, grills, expanded metal, etc., to prevent accidental contact with fan.

d) **Explosion-Proof.** Designed and built to withstand an explosion of gas or vapour within it and to prevent ignition of gas or vapour surrounding machine by sparks, flashes, or explosions which may occur within machine casing.

e) **Dust-Ignition-Proof.** Designed and built to exclude amounts of dust capable of ignition or amounts affecting performance or rating; to prevent ignition of exterior dust on or in vicinity of enclosure.

f) **Waterproof.** Designed to exclude water applied in a stream from a hose. Leakage is permitted around the shaft provided that water is excluded from oil reservoir and means of drainage is provided.

g) **Pipe-ventilated.** Openings so arranged that inlet and outlet ducts or pipes may be connected to them for the admission and discharge of ventilating air.

h) **Water-Cooled.** Cooled by circulating water; the water or water conductors come in direct contact with the machine parts.

i) **Water-Air-Cooled.** Cooled by circulating air which, in turn, is cooled by circulating water.

j) **Air-To-Air Cooled.** Cooled by circulating internal air through heat exchanger which, in turn, is cooled by circulating external air.

MOTOR TYPES

The "family tree" of motor types is quite extensive.

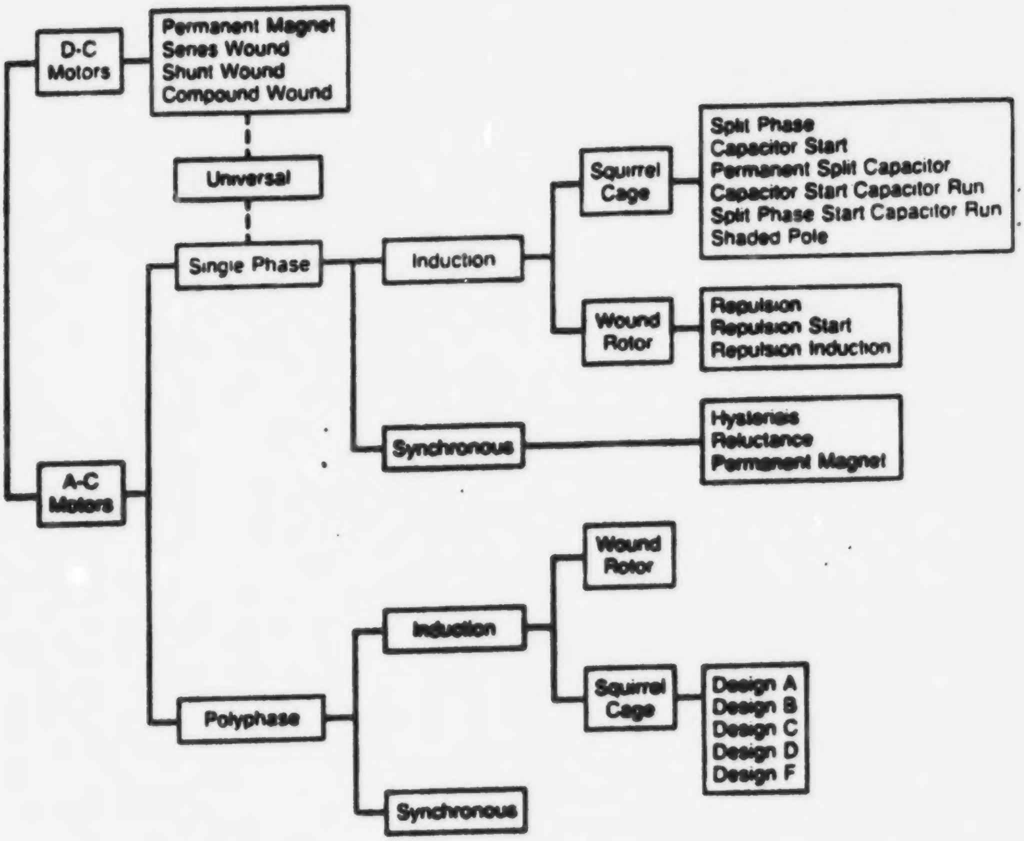


Figure 11-2 Motor Family Tree

Some of the more common types are as follows:

D.C. MOTORS

In a D.C. (direct current) motor the stator (field) winding is supplied by D.C. and the rotor (armature) winding is connected to D.C. by brushes and a commutator. The commutator can be regarded as a switch that maintains the proper direction of current in the rotor conductors to give a constant unidirectional torque.

D.C. motors are often used where a wide range of torque and speed control is required to match the needs of the application. The brush-commutator arrangement, however, presents the problems of brush maintenance and electrical arcing.

When running, the torque produced by a single phase motor is pulsating and irregular, contributing to a much lower power factor and efficiency than for a polyphase motor.

Split Phase Motor

This is the simplest and most common single phase motor. An auxiliary winding produces a phase difference to start the motor and is switched out by a centrifugal switch as running speed is approached.

Capacitor Start Motor

A capacitor in series with the auxiliary winding improves starting torque and reduces starting current. Both the capacitor and the auxiliary winding are switched out by a centrifugal switch.

Permanent Capacitor Motor

The auxiliary winding is permanently connected through a series, low value, continuous duty capacitor. The result is smoother, more efficient running comparing more favourable with three phase operation. This motor has a very low starting torque, sufficient for direct drive fans and blowers but not for belt driven loads. It can also provide speed control for fans (using voltage control).

Capacitor Start/Capacitor Run Motor
(2 Value Capacitor Motor)

This motor has a start capacitor which is switched out after start up to give high starting torque and a permanent capacitor to improve running performance.

Split Phase Start/Capacitor Run Motor

A permanent capacitor in series with the auxiliary winding improves running performance. However, a centrifugal switch by passes the capacitor at start-up to give the starting performance of a split phase motor.

Shaded Pole Motors

This motor produces a low starting torque by using a "shading coil" around part of each pole, rather than using an auxiliary winding. It is a low cost motor, with low efficiency and power factor found in many light duty appliances.

Universal Motor

This motor has brushes and can operate on either D.C. or A.C. It has a very high horsepower/weight ration, handles intermittent overloads easily and can be speed controlled. Its only disadvantage compared to single phase induction motors is a relatively short life. Typically used in vacuum cleaners and lawn mowers.

Synchronous Motors

These motors have A.C. (alternating current) applied to the stator and D.C. applied to the rotor. The result is operation at a constant speed that is fixed by the supply frequency and the number of poles in the motor. The advantage of the synchronous motor is that it is highly efficient and can be made to operate at a leading power factor. Lagging power factor in an industrial plant can be corrected by the use of synchronous motors.

Induction Motors

The induction motor is so named because the A.C. current in the stator "induces" a voltage across an air gap and into the rotor winding.

Squirrel Cage Induction Motor

In this motor the rotor "winding" is essentially a "cage" of conducive bars, short-circuited at the ends. The squirrel cage induction motor is the simplest, most reliable motor owing to the rugged rotor "cage" winding and the absence of brushes. However, speed control is difficult because of the fixed resistance of the rotor cage. Large starting currents required by this motor can cause voltage fluctuations. The general purpose, squirrel cage induction motor (Design B) is truly the "workhorse" of industry.

High Slip Squirrel Cage Induction Motor (Design D)

These are designed to provide high starting torque and low starting current for high inertia loads such as punch presses and well pumping. Strictly speaking, their efficiency is lower than

Design B motors because of higher rotor resistance. Overall system efficiency, however, is usually improved where high inertia loads are involved.

> **NOTE:**
>
> "Slip" is the difference between synchronous speed and running speed, eg. an 1,800 rpm motor, running at 1,745 rpm has a slip of $\left[\frac{1800-1745}{1800}\right] \times 100\% = 3\%$

Wound Rotor Induction Motor

As the name implies, this motor has a wound rotor winding. External connection can be made to this winding through brushes that run on continuous slip rings. Speed and torque can be controlled by connecting an external resistance in series with the rotor winding. Thus, the wound rotor motor is suited to severe starting applications such as ball mills in a mining operation. In comparison, the wound rotor induction motor is not as rugged as the squirrel cage motor but is more reliable than a D.C. motor.

Single Phase Induction Motors

Most single phase induction motors are of the squirrel cage type. While a polyphase motor sets up a rotating field which can start the motor, a single phase motor needs an auxiliary means of starting.

MOTOR MAINTENANCE

The squirrel cage induction motor is a rugged device and, when properly applied, will provide years of service with a minimum of maintenance. There is no question, however, that simple regular maintenance and inspection will provide greater longevity. A record of maintenance history can provide information about when and why motors fail.

Be aware of changes in a motor's unusual performance. For example:

- is the motor noisier than usual?
- is the motor frame temperature higher than normal?
- does the motor take longer to come up to speed?
- is there an increase in vibration?

When properly interpreted these symptoms can help to avert serious trouble or motor failure.

Dirt

If screens or filters become clogged air flow will be reduced and the motor could overheat, eventually leading to insulation failure. Keep the motor clean by wiping with lint-free rags, using suction to draw out dust or clean using compressed air, being careful not to drive dirt further into the machine.

Moisture

Motors that are used continuously are not prone to moisture problems. It is the intermittent use or stand-by motor that may have difficulties. Try to run the motor for at least a few hours each week to drive off moisture. Be careful that steam and water are not directed into open drip-proof motors. Where totally enclosed motors are used in a humid environment remove any condensed water through the drain plug and, if possible, leave the drain plug out.

As a check, "megger" the windings to check the condition of the insulation. The reading for 575 V motors at 25°C should be 1 megohm or greater. If it is not, the windings may require drying.

Greasing

Over-greasing of anti-friction bearings increases friction, causing the bearing to overheat. This is the most common cause of bearing failure. Rather than packing the bearing full of grease until it runs out the plug hole, the bearing should be filled not more than 30%. Then it is still a good idea to leave the drain plug out while the motor runs for about ten minutes, allowing any excess grease to escape.

Be sure not to use contaminated or improper lubricants, as this is another cause of bearing failure.

Vibration

A noticeable increase in motor vibration is an indication of trouble. If it occurs:

- check that all mounting bolts are tight
- check shaft for proper alignment
- bearings may be at the end of their useful life

Identify the source of vibration can sometimes be a difficult problem. Often the motor manufacturer can assist you in measuring and analyzing vibration problems, particularly with large motors.

Starting

Excessive starting is a prime cause of motor failures. The high current flow during start-up contributes a great amount of heat to the motor. For motors 200 hp and below, the maximum acceleration time a motor connected to a high inertia load can tolerate is about 20 seconds. The motor should not exceed more than about 150 "start seconds" per day.

Even with low inertia loads that come up to speed in a fraction of a second, the mechanical stress caused by frequent starting can lead to a premature motor failure, even if the total "start seconds" remain low. Where starting requirements are severe, and with motors 250 hp and above, starting requirements should be discussed with the motor manufacturer prior to purchase.

SUBJECT: TOPIC: 12

PREVENTIVE MAINTENANCE ALIGNMENT

OBJECTIVES:

The trainee will be able to:

1. Recall the problems resulting from faulty alignment.
2. Recall and explain the two types of misalignment.
3. Describe how to conduct:
    a) an angular alignment check
    b) a parallel alignment check
4. Recall the factors that may disturb alignment.

ALIGNMENT

GENERAL

All mechanical equipment where a driving element is involved must be correctly aligned on the job prior to being put into service. This topic will deal only with shaft or coupling alignment. Misalignment can also occur on other units such as those that are belt or chain driven, however correction is much less complex.

Problems resulting from faulty alignment include:

1. noisy operation
2. reduced bearing life
3. excessive coupling wear
4. waste of power

FOUNDATION

To ensure a long-lasting alignment of any unit, the foundation should be sufficiently substantial to absorb any vibration and to form a permanent, rigid support for the base plate. This is important in maintaining the alignment of a direct connected unit. A concrete foundation on a solid base should be satisfactory. Foundation bolts of the proper size should be embedded in the concrete, located by a drawing or template. A pipe sleeve about two and one-half diameters larger than the bolt should be used to allow movement for the final positioning of the bolts.

FACTORY ALIGNMENT

Pumps and drivers that are received from the factory with both machines mounted on a common baseplate, were accurately aligned before shipment. All baseplates are flexible to some extent and, therefore, must not be relied upon to maintain the factory alignment. Preliminary alignment is necessary after the complete unit has been levelled on the foundation and again after the unit is piped, and rechecked periodically as outlined in the following paragraphs.

FIELD ALIGNMENT

The faces of the coupling halves should be spaced far enough apart so that they cannot strike each other when the driver rotor is moved toward the pump. The necessary tools for checking the alignment of a flexible coupling are a straight edge and a taper gauge or a set of feeler gauges. In cases where extreme accuracy is necessary, a dial indicator may be used to align the coupling.

SHAFT ALIGNMENT

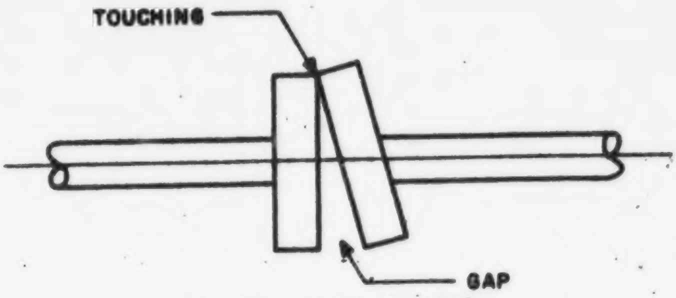
It should be emphasized that when shaft alignment is spoken of, it is shafts and not the couplings which are the subject of discussion. The purpose of the flexible coupling is to compensate for temperature changes and to permit end movement of the shafts without interference with each other while transmitting power from the driver to the pump.

A flexible coupling should not be used to compensate for misalignment of a pump and driver shafts.

Consider, for example, a rigid coupling joining a driven and a driving shaft where the bore of one of the halves of the coupling is not square with the face. Even with the shafts in line, when the half couplings are brought together for bolting, it will be found that whereas at one point the engaging faces touch, at another point a space will exist between these faces, as shown in Figures 12-1 (a).

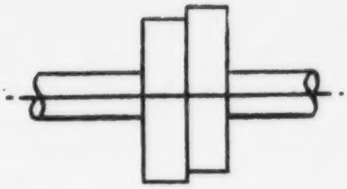
Illustrations 12-1 (a) and 12-1 (b) have been greatly exaggerated for purposes of demonstration.

If under these conditions, the coupling halves are forcibly drawn together by the coupling bolts, the shafts will be forced to take a bowed shape to compensate for the error in the machining of the coupling. Large stresses will be set up in the parts and a pronounced tendency to vibration will be introduced. A situation such as this may be corrected by machining the coupling half on a lathe, so that the coupling face is perpendicular to the shaft axis.



EFFECT OF COUPLING BORE NOT SQUARE WITH COUPLING FACE

Figure 12-1 (a)



EFFECT OF COUPLING NOT CONCENTRIC WITH SHAFT

Figure 12-1 (b)

Thus, it will be seen that in order to effect correct shaft alignment the coupling halves (which are used for aligning the shafts) must have true coupling faces. Generally, the couplings should be checked for trueness of coupling face (square to shaft and flat) and periphery (round and concentric) to the shaft. As a rule the couplings will have been accurately machined and this will not be a problem in alignment. However the installation man should be aware of such a possibility.

The faces of the coupling halves should be spaced far enough apart so that they cannot strike each other when the driver rotor is moved hard over toward the pump. Due allowance should be made for wear of the thrust bearings. A minimum dimension for the separation of the coupling halves is usually specified by the manufacturer. The necessary tools for approximately checking the alignment of a flexible coupling are a straight edge and a taper gauge or a set of feeler gauges.

TYPES OF MISALIGNMENT

There are two types of misalignment, although usually it is a combination of both. Shown in Figure 12-2, and 12-3 these are:

1. **Parallel** or offset misalignment: the shaft axes are parallel but not collinear.
2. **Angular** misalignment: the shaft axes intersect but are not collinear.

Parallel alignment check is made by placing a straight edge across both coupling rims at the top, bottom and at both sides. The unit will be in parallel alignment when the straight edge rests evenly on the coupling rim at all positions. Allowance may be necessary for temperature changes and for coupling halves that are not of the same outside diameter. **Care must be taken** to have the straight edge parallel to the axis of the shafts (See Figure 12-2).

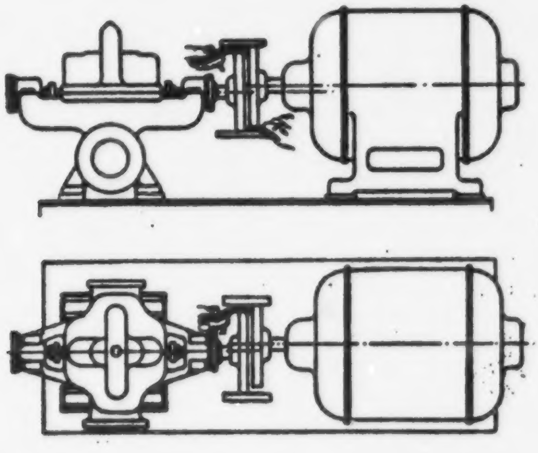


Figure 12-2 Parallel Misalignment

Angular alignment check is made by inserting the taper gauge or feelers between the coupling faces at 90 degree intervals around the coupling. The unit will be in angular alignment when the coupling faces are exactly the same distance apart at all points (See Figure 12 3).

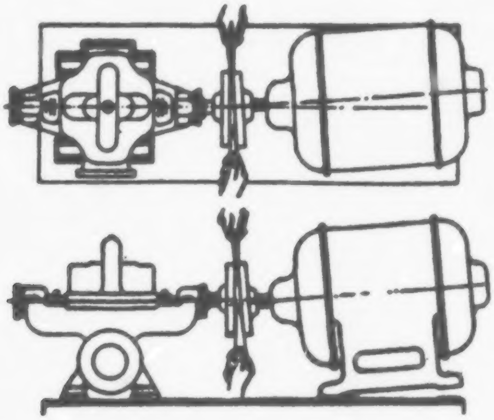


Figure 12-3 Angular Misalignment

Correction for Angular and Parallel Misalignment is made by adjusting the wedges or shims under the baseplate unless pump or motor is mounted in the field. Then after each change, it is necessary to recheck the alignment of the coupling halves. Adjustment in one direction may disturb adjustments already made in another direction.

The permissible amount of coupling misalignment will vary with the type of pump and driver, but in all cases conform to the coupling manufacturer's specifications.

When the preliminary alignment has been completed tighten the foundation bolts and recheck shaft alignment.

If the unit, i.e. pump is to be connected to the prime mover by gears or chains, the alignment should be checked by a straight edge across the faces of the gears or sprockets. This should be done in two directions at an angle as large as permitted by the relative size of the gears or sprockets. When pumps or prime movers will be heated in operation, i.e. pumps handling hot liquids, the unit should be aligned under the thermal conditions so that contraction and expansion due to temperature changes may be taken into account.

ALTERNATE METHOD OF ALIGNMENT

An approved method for putting the coupling halves in final accurate alignment is by the use of a dial indicator. Check alignment by straight edge, taper gauge or feelers as accurately as possible by procedure indicated above.

Bolt the indicator to the pump half of the coupling, with the indicator button resting on the other coupling periphery, set the dial to zero, and chalk mark the coupling half at the point where the button rests. For any check, top or bottom or sides, rotate both shafts by the same amount; i.e. all readings on the dial must be made with its button on the chalk mark.

The dial readings will indicate whether the driver has to be raised or lowered or moved to either side. After each movement, check to see that coupling faces remain parallel to one another.

EXAMPLE: If the dial reading at the starting point is set to zero and the diametrically opposite reading at the bottom or sides shows a plus or minus reading of .020 inch, the driver must be raised or lowered by the use of suitable shims, or, moved to one side or the other by half of this reading.

NOTE: For all checks, including that for parallelism of coupling faces, keep both shafts pressed hard over to one side when taking readings.

ALIGNMENT OF GEAR TYPE COUPLINGS

Gear type couplings are aligned in the same manner as outlined above. However, the coupling covers must be moved back out of the way and measurements made on the coupling hubs as shown on Figure 12 4.

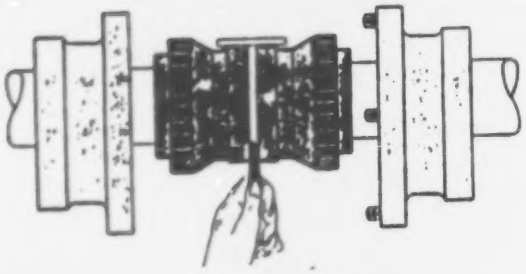


Figure 12-4 Alignment of Gear Type Coupling

GROUTING

Grouting compensates for unevenness in the foundation and prevents vibration and shifting after mounting is complete. Build a form around the baseplate to contain the grout and sprinkle area with water to obtain a good bond. The base should be completely filled with a good quality non shrinking grout. The usual mixture for grouting is one part Portland cement and two parts sand with sufficient water to flow freely. It is also desirable to grout the levelling pieces, shims or wedges in place. Foundation bolts should be tightened when grout has hardened, usually about 48 hours after pouring.

FINAL CHECK OF ALIGNMENT

After the grout has set and the foundation bolts have been properly tightened, the unit should be checked for parallel and angular alignment and, if necessary, corrective measures taken.

After the piping of the unit has been connected, the alignment should be again checked.

The direction of rotation of the driver should be checked to make certain that it matches that of the pump. The corresponding direction of rotation of the pump is usually indicated by a direction arrow on the pump casing.

The coupling halves can then be reconnected. With the pump properly primed, the unit then should be operated under normal operating conditions until temperatures have stabilized. It then should be shut down and immediately checked again for alignment of the coupling. All alignment checks must be made with the coupling halves disconnected and again after they are connected.

It should be emphasized that attempts to correct alignment in one direction may alter the alignment in the other direction; therefore, it is necessary to check in all directions after making any adjustments.

After the unit has been running for about one week, the coupling halves should be given a final check for misalignment caused by pipe strains or temperature strains. If the alignment is correct, both pump and driver should be dowelled to the baseplate. The location of the dowels is very important and the manufacturer's instructions should be obtained, especially if the unit is subject to temperature changes.

FACTORS AFFECTING ALIGNMENT

The unit should be checked periodically for alignment. If the unit does not stay in line after being properly installed, the following are possible causes:

1. Settling, seasoning or springing of the foundation.
2. Pipe strains distorting or shifting the machine. Both suction and discharge pipes should be supported independently near the pump so that when the flange bolts are tightened no strain will be transmitted to the pump casing.
3. Wear of the bearings.
4. Springing of the base plate by heat from an adjacent steam pipe or from a steam turbine.
5. Shifting of the building structure due to variable loading or other causes.
6. It may be necessary to slightly re-adjust the alignment, from time to time, while the unit and foundation are new.

SUBJECT: **TOPIC: 13**

PREVENTIVE MAINTENANCE **MOTOR STARTER**

OBJECTIVES:

The trainee will be able to:

1. Understand the operating characteristics of various starters.
2. Understand the working of a manual starter and a magnetic starter.
3. Understand how to select manual and magnetic starters.

MOTOR STARTERS

Motor starters can be anything from a simple toggle switch to a complex system with solenoids, relays, and timers. Their function is simply to control the operation of a motor. Motor control includes means for starting, stopping, accelerating, and decelerating motor speeds. A specific control scheme may do only one, some, or all of these functions.

Most A-C motors are started directly across the supply lines. To do this successfully, a motor must have windings capable of taking a sudden mechanical stress, accelerate fast enough to prevent overheating, and hold the inrush current limits established by the power company. There are two basic across-the-line motors starters: the simple manual starter and the magnetic starter.

Selection of Motor Controls

The motor, machine, and motor controller are interrelated and need to be considered as a package when choosing a specific device for a particular application. The motor should be matched to the electrical service, and correctly sized for the machine load (horsepower rating). Other considerations include motor speed and torque. To select proper protection for the motor, its **FULL-LOAD CURRENT** rating, service factor, and time rating must be known. These are obtained from the motor's nameplate.

Motor control equipment is designed to meet the provisions of the National Electrical Code (NEC). Code sections applying to industrial control devices are Article 430 on motors and motor controllers and Article 500 on hazardous locations.

Standards established by the National Electrical Manufacturers Association (NEMA) help users select proper control equipment. Underwriters' Laboratories (UL) is an organization that actually tests for conformity to national codes and standards.

Motor Controllers

The fundamental job of a motor controller is to start and stop the motor, and protect the motor, machine, and operator.

A motor controller might also be called on to provide supplemental functions such as reversing, changing speed, jogging or inching, plugging, sequence control, overload protection, over-current protection, and pilot light indication. The controller can also provide control for auxiliary equipment, such as brakes, clutches, solenoids, heaters, and signals. A motor controller can be used to control either a single motor or a group of motors.

Controller Enclosures

Controller enclosures provide protection for operating personnel by preventing accidental contact with live parts. In certain applications, the controller itself must be protected from a variety of environmental conditions such as:

1. Water, rain, snow, or sleet
2. Dirt or noncombustible dust
3. Cutting oils, coolants, or lubricants

Both personnel and property require protection in environments made hazardous by the presence of explosive gases or combustible dusts.

The classifications for enclosures are:

NEMA 1 **GENERAL PURPOSE** enclosure that protects against dust and light indirect splashing of water; it is not dust tight.

NEMA 2 **DRIP-TIGHT** enclosure suitable for applications where moisture condensation may be severs; encountered in cooling rooms and laundries.

NEMA 3	**WEATHER-RESISTANT** enclosure suitable for outdoor application. It is splash proof, weatherproof, sleet-proof, and moisture-resistant.
NEMA 4	**WATERTIGHT** enclosure suitable for outdoor applications and in dairies, breweries, and other wet locations.
NEMA 5	**DUST-TIGHT** enclosure. It is no longer supplied by most manufacturers. NEMA 12 is substituted.
NEMA 7	**HAZARDOUS LOCATIONS - CLASS I** enclosure for locations where flammable gases or vapours exist or may be present in the air.
NEMA 9	**HAZARDOUS LOCATIONS - CLASS II** enclosure for locations where combustible dust is present.
NEMA 12	**INDUSTRIAL USE** enclosure. It is designed to exclude dust, lint, oil, and coolant seepage. There are no conduit openings or knock-outs in the enclosure; you must punch your own as required.

Starters

The terms "starter" and "controller" mean practically the same thing. Strictly speaking, a starter is the simplest form of controller and is capable of starting and stopping the motor and providing it with overload protection.

Manual Starters

A manual starter as illustrated in figure 13-1 is a motor controller whose contact mechanism is operated by a mechanical linkage from a toggle handle or push-button that is operated by hand. A thermal unit and direct-acting overload mechanism provides motor running overload protection. Basically, a manual starter is an "ON-OFF" switch with overload relays.

Manual starters are used on small machine tools, fans and blowers, pumps, compressors, and conveyors. They are simple and they provide quiet operation. Moving a handle, or pushing the START button, closes the contacts, which remain closed until the handle is moved to "off", or the STOP button is pushed, or the overload relay thermal units trip.

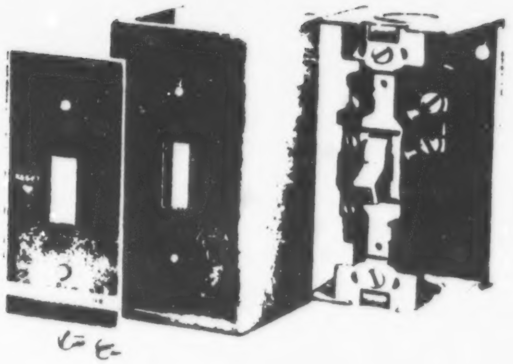
Manual starters are of the fractional or the integral-horsepower type and usually provide across-the-line starting. Manual starters cannot provide low voltage protection or low voltage release. If power fails, the contacts remain closed, and the motor will restart when power returns. This is an advantage for pumps, fans, compressors, and oil burners. But for other applications it can be a disadvantage and can be dangerous to personnel or equipment.

Manual Motor Starting Switches

Manual motor starting switches provide on/off control of single phase or 3-phase A-C motors where overload protection is not required, or is provided separately. Two-pole or 3-pole switches are available with ratings up to 5 hp, 600 volt, 3-phase. The continuous current rating is 30 amperes at 250 volts maximum, and 20 amperes at 600 volts maximum.

Integral-horsepower manual starters are available in pole and 3-pole versions to control single-phase motors up 5 hp and 3-phase motors up to 10 hp.

Most 2-pole starters have one overload relay; 3-pole starters usually have two overload relays but are available with three overload relays. When an overload relay trips, the starter mechanism unlatches, opening the contacts to stop the motor. The contacts cannot be reclosed until the starter mechanism has been reset by pressing the STOP button or moving the handle to the RESET position after the thermal unit has cooled.



Square D Co

Figure 13-1 A Single-Phase Manual Starter

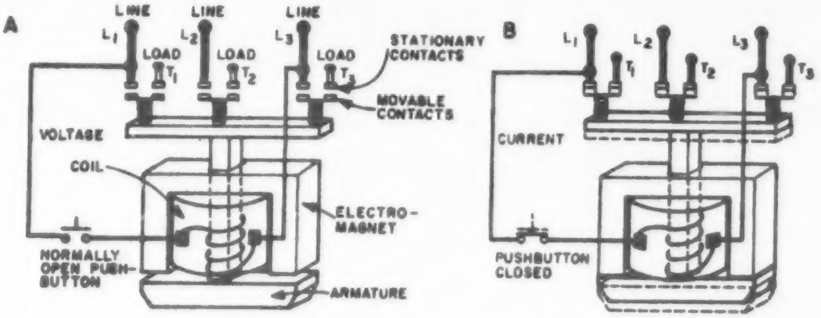


Figure 13-2 Magnetic Starter Operation

Magnetic Controls

A high percentage of applications require the controller to be operated from remote locations, or to operate automatically in response to signals from such pilot devices as thermostats, pressure or float switches, and limit switches. Low voltage release or protection may also be desired. Manual starters cannot provide this type of control. Therefore, magnetic starters are necessary.

The use of an electromagnet distinguishes a magnetic starter from a manual starter. An electromagnet consists of a coil of wire placed on an iron core as illustrated in Figure 13.2 (a). When current flows through the coil, the iron of the magnet becomes magnetized, attracting the iron bar called the armature, as illustrated in Figure 13.2 (b).

Armature Assemblies

In the construction of a magnetic controller, the armature of the electromagnet is mechanically connected to a set of contacts. When the armature moves to its closed position, the contacts also close. When the coil has been energized, and the armature has moved to the closed position, the controller is said to be "picked up", and the armature "sealed" or "sealed-in".

Types of armature assemblies are:

CLAPPER - The armature is hinged. As it pivots to seal in, the movable contacts close against the stationary contacts, as illustrated in Figure 13-3 (a).

VERTICAL ACTION - The action is a straight line motion with the armature and contacts being guided so that they move in a vertical plane as illustrated Figure 13-3 (b).

BELLCRANK - A bellcrank lever transforms the vertical action of the armature into a horizontal contact motion. The shock of armature pickup is not directly transmitted to the contacts, resulting in minimum contact bounce and longer contact life, as illustrated in Figure 13-3 (c).

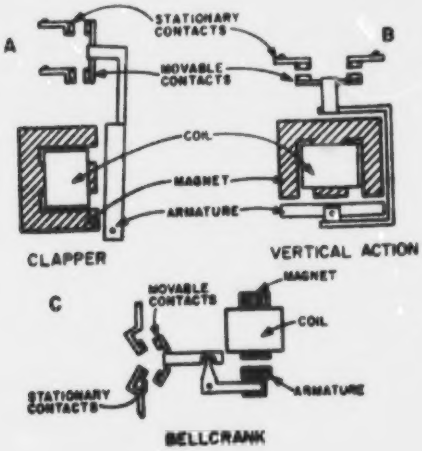


Figure 13-3 (a, b & c) Starter Armature Assemblies.

Magnetic Circuits

The magnetic circuit of a controller consists of the magnet assembly, the coil, and the armature. The coil, and the current flowing in it, cause a magnetic flux to be set up through the iron. The changing magnetic flux produced by A-C results in a temperature rise in the magnetic circuit, which is not desirable. Therefore the heating effect is reduced by laminating the magnet assembly and armature.

The magnet assembly is the stationary part of the magnetic circuit. The coil is supported by and surrounds part of the magnet assembly to induce magnetic flux in the magnetic circuit.

When the armature has been attracted into its sealed-in position, it completes the magnetic circuit. To provide maximum pull (to close the contacts), the magnet assembly is machined to a very close tolerance.

When a controller's armature has sealed-in, it is held closely against the magnet assembly. However, a small gap is always deliberately left in the iron circuit, as illustrated in Figure 13-4. When the coil is de-energized, some magnetic flux (residual magnetism) always remains, and if it were not for the air gap in the iron circuit, the residual magnetism would eventually become sufficient to hold the armature in the sealed in position.

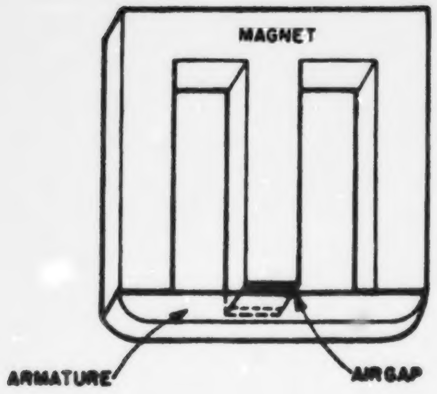


Figure 13-4 Magnet Assembly with Air Gap

Shading Coil

A shading coil is a single turn of copper or aluminum mounted in the face of the magnet assembly or armature, as illustrated in Figure 13-5. The alternating main magnetic flux induces a voltage in the shading coil, causing currents to set up an auxiliary magnetic flux that is out of phase from the main magnetic flux. The auxiliary magnetic flux produces a magnetic pull out of phase with the pull of the main magnetic flux and this keeps the armature sealed-in when the main magnetic flux falls to zero (120 times per second with 60 hz). Without the shading coil, the armature would tend to open each time the main flux goes through zero. Excessive noise, wear on the magnet faces, and heat would result.

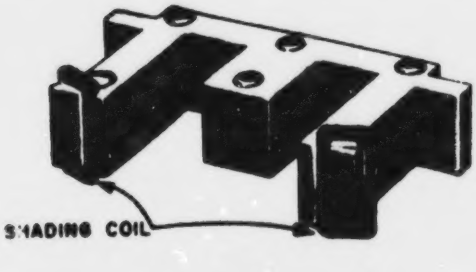


Figure 13-5 Shading Coils

Magnetic Coils

Magnetic coils have many turns of insulated copper wire wound on an insulated spool. Most coils are protected by an epoxy moulding, which makes them resistant to mechanical damage.

When a controller is in the open position there is a large air gap (not to be confused with the built-in air gap) in the magnetic circuit because the armature is at its farthest distance from the magnet. The impedance of the coils is relatively low because of the air gap. Therefore, when the coil is energized, it draws a high current. As the armature moves closer to the magnet assembly, the air gap is progressively reduced. The coil current decreases as the air gap is reduced until the armature has sealed-in. The final coil current is referred to as the sealed current. The in-rush current is approximately 6-to-10 times the sealed current. (The ration varies with individual designs.) After the controller has been energized for some time, the coil becomes hot. This causes the coil current to fall to approximately 80 percent of its value when cold.

A-C magnet coils of controllers should never be connected in series. The increased circuit impedance reduces the coil current. If one device were to seal-in ahead of the other, the slower device would not pick up, or, having picked up, would not seal. Therefore, A-C contactor coils must always be connected in parallel.

Magnet coil data is usually given in volt amperes (VA). For example, given a magnetic starter where coils are rated at 600 VA inrush and 60 VA sealed; the inrush current of a 120 volt coil is 600/120 or 5 amperes, and the sealed current is 60/120 or 0.5 amperes. The same starter with a 480 volt coil has an inrush current of 600/480 or 1.25 amperes and 60/480 or 0.125 amperes sealed.

The minimum control voltage, which will cause the armature to start to move, is called the pick-up voltage.

The seal-in voltage is the minimum control voltage required to cause the armature to seat against the pole faces of the magnet. In devices using vertical action magnet and armature, the seal-in voltage is higher than the pick-up voltage to provide additional magnetic pull for good contact pressure.

Control devices using the bellcrank armature and magnet arrangement have different force characteristics than the vertical action armature. Devices using the bellcrank operating principle are designed to have a lower seal-in voltage than pick-up voltage. This extends contact life and contact damage and abnormal voltage conditions is reduced. If the voltage is sufficient to pick-up, it is also high enough to seat the armature.

If the control voltage is reduced sufficiently, the controller will open. The voltage at which this happens is called the drop-out voltage and is always lower than the seal in voltage.

Effects of Voltage Variation

If the voltage applied to the coil is too high, the coil will draw more current than its rated current. Excessive heat is then produced, which causes early failure of the coil insulation. The magnetic pull becomes too high, and causes the armature to slam home with excessive force. The magnet faces wear rapidly, leading to a shortened life for the controller. The contact bounce may also be excessive, resulting in reduced contact life.

Low control voltage produces low coil currents and reduced magnetic pull. On devices with vertical action assemblies, if the voltage is greater than pick up voltage but less than seal-in voltage, the controller may pick up but will not seal. With this condition, the coil current will not fall to the sealed value. Because the coil is not designed to carry a current greater than its sealed current continuously, it will quickly get very hot and burn out. The armature will chatter. In addition to the noise, wear on the magnet faces results.

In both the vertical action and bellcrank construction, if the armature does not seal, the contacts will not close with adequate pressure. Excessive heat, with arcing and possible welding of the contacts, then occurs as the controller attempts to carry current with insufficient contact pressure.

Magnetic Starters - NEMA Sizes and Ratings

Power circuit contacts of a starter carry the MOTOR LOAD. The ability of the contacts to carry the full-load current without exceeding a rated temperature rise, and their isolation from adjacent contacts, correspond to NEMA Standards established to categorize the NEMA size of the starter. The starter must also be capable of interrupting the motor circuit under locked-rotor current conditions.

Table 1 Starter Sizes

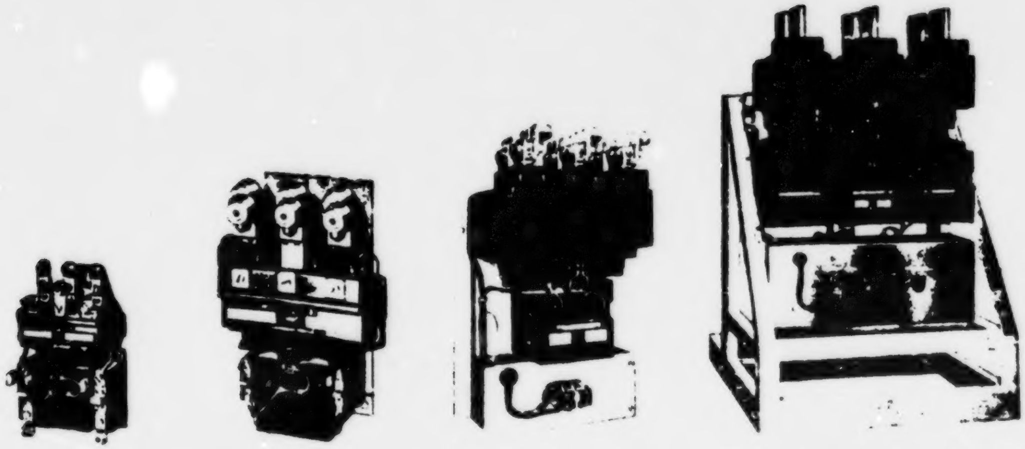
SEPARATE CONTROL – 120 VOLT									
NEMA Size	Continuous Ampere Rating	Maximum Horsepower Rating Full load current must not exceed "Continuous Ampere Rating"			NEMA Size	Continuous Ampere Rating	Maximum Horsepower Rating Full load current must not exceed "Continuous Ampere Rating"		
		Volts	Single Phase	3 Phase			Volts	Single Phase	3 Phase
00	9	208	1	1½	5	270	208	–	100
		240	1	1½			240		100
		480	–	2			480		200
		600	–	2			600		200
0	18	208	2	3	6	540	208	–	200
		240	2	3			240		200
		480	–	5			480		400
		600	–	5			600		400
1	27	208	3	7½	7	810	208	–	300
		240	3	7½			240		300
		480	–	10			480		600
		600	–	10			600		600
2	45	208	7½	15	8	1215	208	–	450
		240	7½	15			240		450
		480	–	25			480		900
		600	–	25			600		900
3	90	208	15	30	9	2250	208	–	800
		240	15	30			240		800
		480	–	50			480		1600
		600	–	50			600		1600
4	135	208	–	50					
		240		50					
		480		100					
		600		100					

A magnetic starter must be equal to or exceed the motor horsepower and full-load current ratings. For example: what size full voltage starter is required for a 100 hp, 480 volt, 125 amp, 3-phase motor? Referring to Table 1, column heading, **Continuous Ampere Rating**, locate 135, the closest rating to the 125 amperes of the motor. The size of the starter is 4 in the column, **NEMA Size**. In the column, **Maximum Horsepower Rating**, there are three sub-columns; **Volts**, **Single-Phase** and **3-Phase**. For a NEMA 4 starter at 480 volts 3-phase, the maximum horsepower rating is 100 hp.

The heading, **Separate Control 120 Volt**, means the coil for the electromagnet is operated at 120 volts. The control circuits for starters are usually operated at a lower voltage than the motor operating voltage. This is a safety precaution. Operating personnel come in contact with push-buttons or other pilot devices which will be only 120 volts to ground rather than 480 volts.

There are eleven standard NEMA starter sizes from Size 00 at 9 amperes, 480 volts 2 ph to Size 9 at 2,250 amperes, 480 volts, 1,600 hp. Starters are also rated for 600 volts maximum, 60 hz. Figure 13-6 shows the relative physical size of four NEMA size starters.

The most popular starter is NEMA size 1. It is used for motors from 2 hp to 10 hp on 480 volts. Sizes 00 and 0 are seldom used. Only replacement parts for one size need be stocked rather than three.



SIZE 3 SIZE 5 SIZE 7 SIZE 9

Figure 13-6 Relative Physical Sizes of Four Starters

A-C Hum

All A-C devices that incorporate a magnetic effect produce a characteristic hum. This noise is caused by the changing magnetic pull (as the magnetic flux changes) which includes mechanical vibration. Contactors, starters, and relays can become excessively noisy as a result of any of the following operating conditions:

1. Broken shading coil.
2. Operating voltage too low.
3. Wrong coil.
4. Misalignment between the armature and magnetic assembly the armature is then unable to seat properly.
5. Dirt, rust, and filings on the magnet faces - the armature is unable to seal-in completely.
6. Jamming or binding of moving parts (contacts, springs, guides, yoke bars) so that full travel of the armature is prevented.
7. Incorrect mounting of the controller, as on a thin piece of plywood fastened to a wall, for example, so that a "sounding board" effect is produced.

Magnetic Starters - Control Circuits

The circuit to the magnet coil, which causes a magnetic starter to pick up and drop out, is distinct from the power circuit as illustrated in Figure 13-7. Although the power circuit can be single phase or 3-phase, the coil circuit is always a single-phase circuit. Elements of a coil circuit include the following:

1. The magnet coil.
2. The contacts of the overload relay assembly.

3. A momentary or maintained-contact pilot device, such as a push button station, pressure, temperature, liquid level, or limit switch.

4. In lieu of a pilot device, the contacts of a relay or timer.

5. An auxiliary contact on the starter, designated as a holding circuit interlock, is required in certain control schemes.

The coil circuit is generally identified as the CONTROL CIRCUIT, and contacts in the control circuit carry the COIL LOAD.

Holding Circuit Interlocks

The holding circuit interlock on standard magnetic starters and contactors is normally open (NO) auxiliary contact. It closes when the coil is energized to form a holding circuit for the starter after the START button has been released.

Square D Co

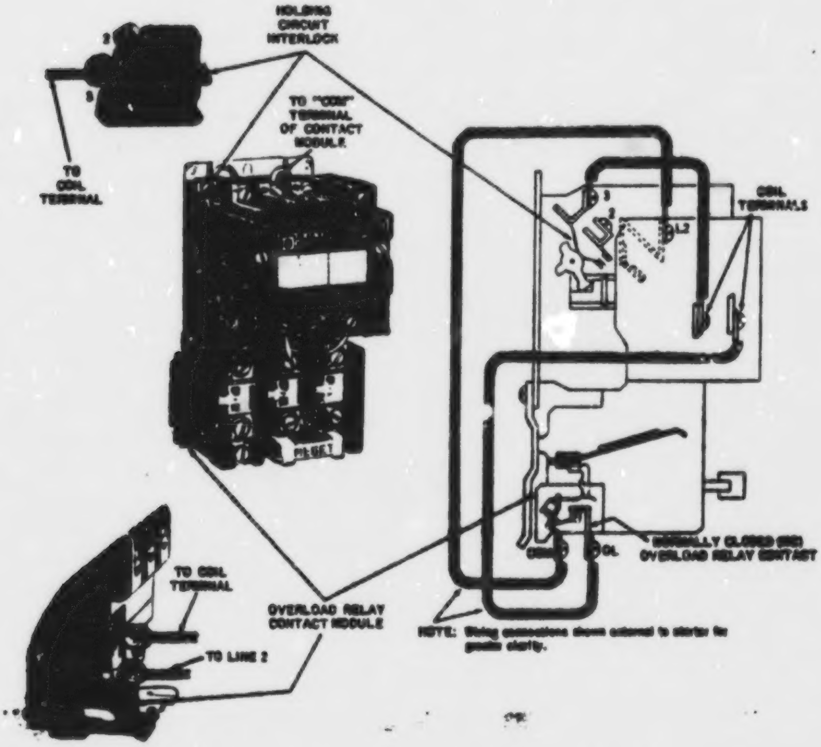
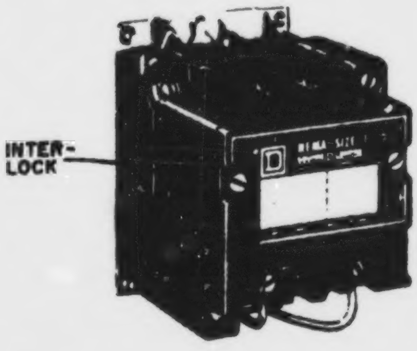

HOLDING CIRCUIT INTERLOCK
2
3
TO COIL TERMINAL
TO "COM" TERMINAL OF CONTACT MODULE
3
2
L2
COIL TERMINALS
RESET
NORMALLY CLOSED (NC) OVERLOAD RELAY CONTACT
OL
TO COIL TERMINAL
TO LINE 2
OVERLOAD RELAY CONTACT MODULE
NOTE: Wiring connections shown external to starter for graphic clarity.

Figure 13-7 A Magnetic Starter Control Circuit



Square D Co

Figure 13-8 Externally Attached Electrical Interlocks

Electrical Interlocks

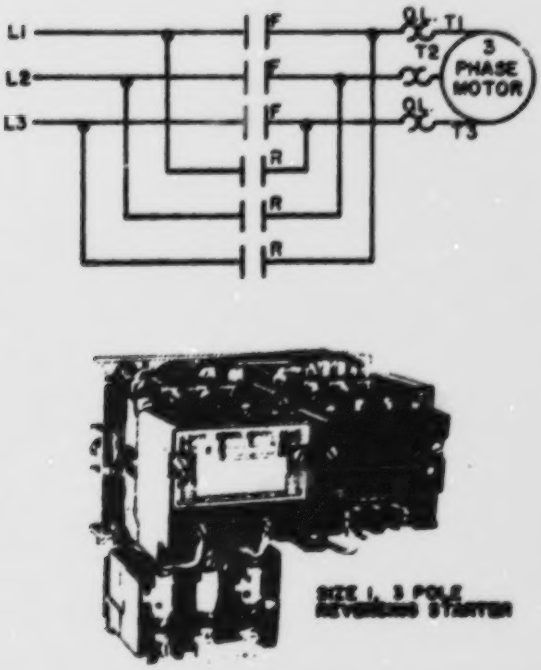
In addition to the main or power contacts, which carry the motor current, and the holding circuit interlock, a starter with externally attached auxiliary contacts called electrical interlocks can be provided, as illustrated in Figure 13-8. Interlocks are rated to carry only control circuit current, not motor current. NO or NC contacts are available.

Interlocks can be used to control other magnetic devices where sequence operation is desired: to prevent another controller from being electrically energized at the same time; and to make and break circuits to indicating devices, such as pilot lights, bells, or other signals. Electrical interlocks are packaged in kit form so they can be easily added in the field.

Reversing Starters

Reversing the direction of motor shaft rotation is often required. Three-phase motors can be reversed by reconnecting any two of the three line connections to the motor. The reconnection can be made by interwiring two contactors.

In the power circuit illustrated in Figure 13-9, the contacts of the FORWARD contactor, when closed, connect Lines 1, 2 and 3 to motor terminals T1, T2 and T3 respectively. As long as the FORWARD contacts are closed, mechanical and electrical interlocks prevent the REVERSE contactor from being energized.



Square D Co

Figure 13-9   A Reversing Magnetic Starter

When the FORWARD contactor is de-energized, the second contactor can be picked up, closing its contacts, R, which reconnect the lines to the motor. Note that by running through the REVERSE contacts, Line 1 is connected to motor terminal T3, and Line 3 is connected to motor terminal T1. The motor will now run in the opposite direction.

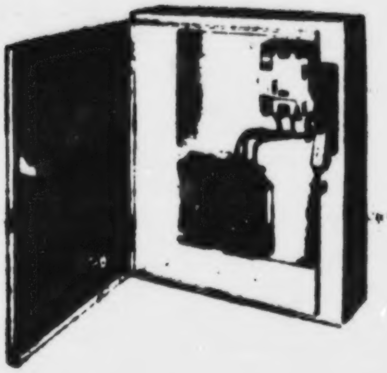
Whether operating through either the forward or reverse contactor, the power connections are made to an overload relay assembly, which provides motor overload protection. A magnetic reversing starter consists of a starter and contactor, suitably interwired, with electrical and mechanical interlocking, to prevent the coil of both units from being energized at the same time.

Combination Starters

Combination starters combine a disconnect means and a magnetic starter in one enclosure, as illustrated in Figure 13-10. Compared to a separately mounted disconnect and starter, the combination starter takes up less space, requires less time to install and wire, and provides greater safety.

Safety to personnel is ensured because the door is mechanically interlocked so it cannot be opened without first opening the disconnect.

Combination starters can be furnished with either circuit breakers or fuses to provide over-current protection and are available in reversing and non-reversing versions.



Allen Bradley Co

Figure 13-10 A Combination Starter

SUBJECT:

PREVENTIVE MAINTENANCE

TOPIC: 14

ELECTRICAL MEASURING INSTRUMENTS

OBJECTIVES:

The trainee will be able to:

1. Understand and identify the use of Ohmmeter, Voltmeter and Ampmeter.
2. Have hands-on working experience and operate the instruments.
3. Understand the safe way to handle instruments demonstrated.

RESISTANCE MEASUREMENTS

There are two general methods of measuring resistance values: 1) directly, by the use of an ohmmeter or megohmmeter, and 2) indirectly, by use of a wheatstone bridge, or an ammeter and voltmeter, and then applying Ohm's Law. Both methods require a source of current, though the method applied depends on the range of measurement and the required accuracy.

The ohmmeter is used to measure resistance and check the continuity of electrical circuits and devices. Its range usually extends to only a few megohms. The megohmmeter is used for measuring insulation resistance of wires, as between the wire and the outer surface of its insulation, and the insulation resistance of cables and insulators. The range of a megohmmeter may extend to more than 1,000 megohms.

The principles and methods of resistance measuring, which are conducted on a day-to-day basis in an industrial plant, are explained in this lesson.

Ohmmeter Operating Principles

An ohmmeter is used to measure resistance and uses a D'Arsonval (PM-MC) movement. An ohmmeter's movement is usually very sensitive, and it requires only a small amount of current for full-scale deflection.

An ohmmeter consists of a D-C milli-ammeter type movement with the following added features:

1. A D-C source of potential (usually a small battery).
2. One or more resistors (one of which is variable).

The simple ohmmeter circuit illustrated in Figure 14-1 shows the basic circuit of the ohmmeter, which consists of a meter movement, a current limiting resistor, a zero adjustment resistor, a meter mechanism for the meter movement, and a battery. Full-scale deflection is obtained when external resistance is being measured with the meter leads shorted together. Therefore, the current limiting resistor of the meter circuit must limit the current to the value for which the movement is rated.

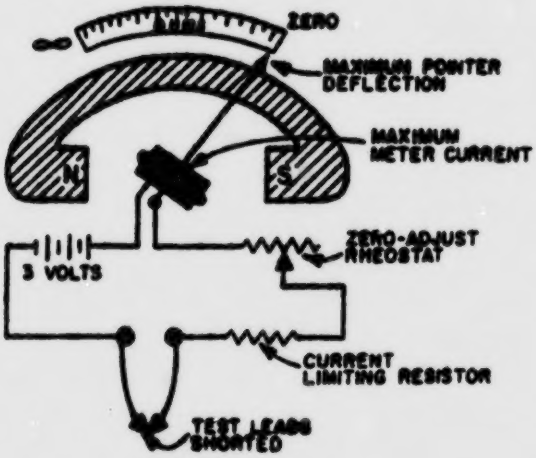


Figure 14-1 Ohmmeter Circuit With Shorted Test Leads

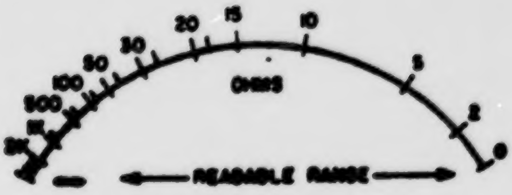


Figure 14-2 Single Range Ohmmeter Scale

The pointer deflection of the ohmmeter is controlled by the amount of battery current that passes through the moving coil. Before you measure the unknown resistance of a resistor or an electrical circuit, the test leads of the ohmmeter should be shorted together as illustrated in Figure 14-1, to adjust the meter pointer to zero.

Ohmmeter Calibration

An ohmmeter is calibrated by shorting the leads together and setting the pointer to zero by means of the variable resistor as follows:

First, set the ohmmeter to the selected range, because the zero setting usually varies for different ranges. Short the leads together. (While the leads are shorted together, current flow is maximum and the pointer is deflected full-scale). Use the variable resistor to bring the pointer to rest exactly on the zero graduation. This indicates **ZERO RESISTANCE** between the shorted test leads.

NOTE: The zero readings of ohmmeters are usually on the right-hand side of the scale, whereas the zero readings for ammeters and voltmeters are generally on the left-hand side of the scale.

The scale of an ohmmeter, Figure 14-2, is known as a reversed scale because the zero graduation is on the right, which corresponds to full-scale deflection. That is, maximum deflection of the meter indicates zero ohms of resistance. The scale is calibrated relative to battery current and total resistance - total resistance being the resistance of the meter plus the resistance being measured.

For example, use a basic ohmmeter circuit as illustrated in Figure 14-3, then add a resistance (dashed lines in the illustration) equal to that of the meter across the terminals. Then determine what value the meter will indicate on the scale. To do this, first calculate the current through the meter. If the battery is 1.5 volts and the total resistance is 3,000 ohms, apply Ohm's Law to arrive at the answer of 500 microamps. This is half the full-scale current, so the meter will register half-scale deflection. This point is marked on the scale in ohms rather than amperes. The other points on the scale are also marked in a similar manner.

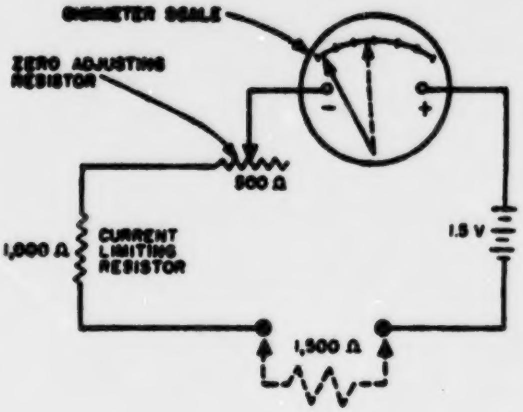


Figure 14-3   Ohmmeter Circuit Example

Note that the graduations on the ohmmeter scale are not evenly spaced (Figure 14-2), and that the pointer deflection becomes less as the resistance increases. This occurs because the greater the resistance is, the smaller the value of current flow for any given voltage. This is a simple application of Ohm's Law.

CAUTION:

NEVER use an ohmmeter to measure resistance of an energized circuit. The internal circuit is designed to carry only the current produced by its own battery. Its voltage is usually between 1.5 and 9 volts. Any voltage from an external source will likely be larger than this value and will damage either the meter coil, the pointer or both - usually beyond repair.

Use of the Ohmmeter

A multirange ohmmeter with the zero adjust and range selector is illustrated in Figure 14-4. Each time the meter is to be used for reading ohms, and each time it is switched to a new range, short the test probes together and zero the pointer with the zero ohms adjustment control (the variable resistor or potentiometer).

After the ohmmeter is adjusted for a zero-ohms reading, it is ready to be connected into a de-energized circuit to measure resistance. A typical ohmmeter circuit arrangement is illustrated in Figure 14-5.

Figure 14-4   A Multirange Ohmmeter

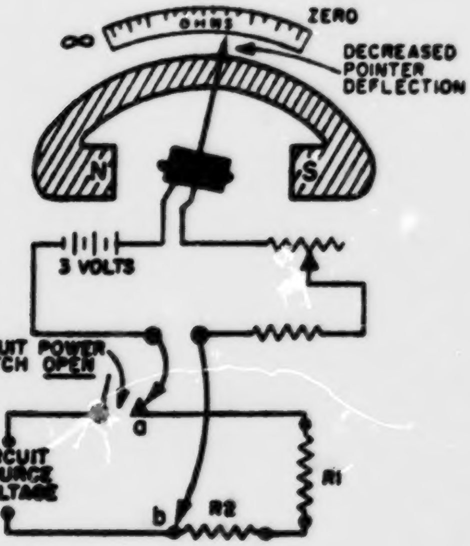


Figure 14-5   Ohmmeter Circuit for Measuring Resistance

The test leads of the ohmmeter are connected across the circuit to be measured. This causes the current produced by the meter's battery to flow through the circuit being tested. Assume that the meter test leads are connected at points A and B of Figure 14-5. The amount of current that flows through the meter coil will depend on the resistance of the meter. Because the resistance of the meter has been adjusted, the amount of coil movement now depends solely on the resistance of R1 and R2. The resistance of R1 and R2 raises the total series resistance in the circuit, decreases the current, and thus decreases the pointer deflection away from zero. The pointer now comes to rest at a scale figure that indicates the combined resistance of R1 and R2. If R1 and R2 were replaced with a resistor that has a large ohmic value, the current flow in the moving coil of the meter would be decreased still more. The deflection would also be further decreased, and the scale indication would read a still higher circuit resistance. Movement of the moving coil is proportional to the amount of current flow. The scale reading of the meter, in ohms, is inversely proportioned to the current flow in the moving coil.

Ohmmeter Multiplication Features

The amount of circuit resistance to be measured may vary over a wide range. In some cases the amount may be only a few ohms, and in other cases, the amount may be as great as one million ohms. To enable the meter to measure values with the least error, multiplication features are incorporated. For example, an ohmmeter may have four test lead jacks, marked as follows: Common: R x 1, R x 10, and R x 100. The jack marked Common is connected internally through the battery to one side of the moving coil of the ohmmeter. The jacks marked R x 1, R x 10, and R x 100 are connected to three resistors of different resistance values located within the ohmmeter. This circuit is illustrated in Figure 14-6.

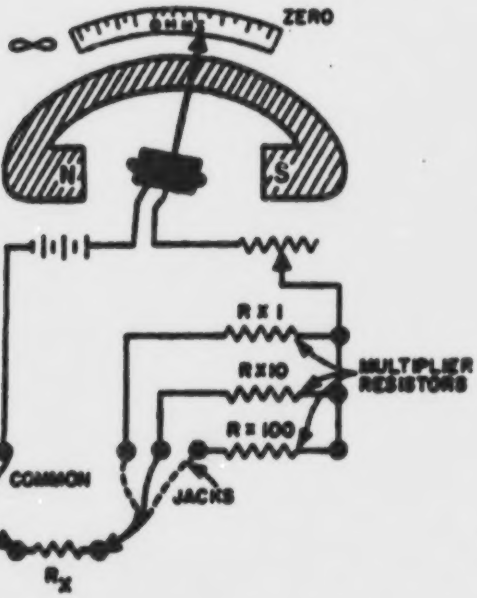


Figure 14-6 Multiple Range Ohmmeter Circuit for Measuring Resistance

Ohmmeters may also be equipped with a selector switch for selecting the multiplier scale desired, so that only two test lead jacks are necessary, usually marked **COMMON** and **OHMS**. Other meters have a separate jack for each range. The range to be used in measuring a particular unknown resistance depends on the approximate ohmic value of the unknown resistance. For example, assume the ohmmeter is calibrated in divisions from zero to 1,000. If R_x is greater than [illegible] ohms and the R x 1 range is being used, the ohmmeter cannot properly measure it and the pointer just barely deflects. The combined series resistance of resistor R x 1 and R_x is too great to allow sufficient battery current to flow to deflect the pointer very much away from infinity (∞). The test lead should be plugged into the next range, R x 10. With this done, the pointer deflects to a readable value, for example: 375 ohms. This indicates that R_x has 375 ohms times 10, or 3,750 ohms. The change in range increased pointer deflection because resistor R x 10 has only one-tenth the resistance of the resistor R x 1. Selecting the smaller series resistance, thus permitted a sufficient amount of battery current to cause a readable pointer deflection. If the R x 100 range were used to measure the same 3,750 ohm resistor, the pointer would deflect still further, to the

37.5 ohm position. This increased deflection would occur because resistor R x 100 has only one-tenth the resistance of resistor R x 10.

It always takes the same amount of current to deflect the pointer to a certain position on the scale (mid-scale position, for example), regardless of the multiplication factor being used. Because the multiplier resistors are of different values, it is always necessary to zero adjust the meter for each multiplication factor when you select it. Try to select the multiplication factor that results in the pointer coming to rest as near as possible to the midpoint of the scale. This will enable you to read the resistance more accurately because the scale readings are more easily interpreted at, or near, midpoint.

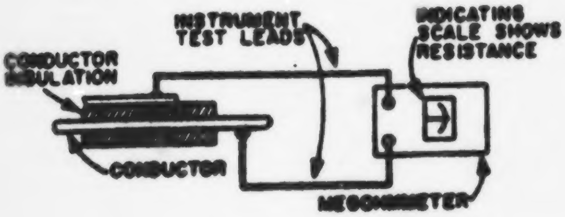


Figure 14-7 Connections for Measuring Insulation Resistance

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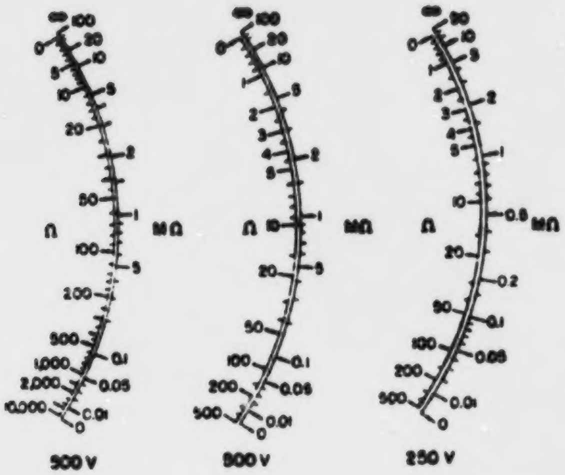


Figure 14-8 Typical Megohmmeter Scales

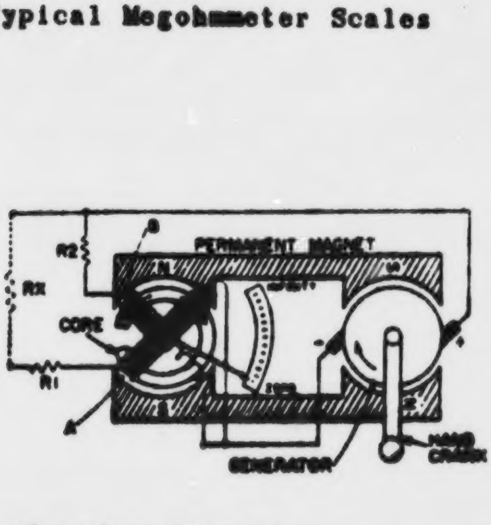


Figure 14-9 Megohmmeter Circuitry

The Megohmmeter

An ordinary ohmmeter cannot be used for measuring resistance of many millions of ohms, such as resistance of conductor insulation. To properly test for "insulation breakdown", it is necessary to use a much higher potential than is furnished by an ohmmeter's battery. This potential is placed between the conductor and the outside surface of the insulation, as illustrated in Figure 14-7.

A megohmmeter is used for these tests. A megohmmeter is frequently called a **Megger** (Trade Mark). It is a portable instrument consisting of; 1) a hand-driven D-C generator, which supplies the necessary voltages for taking the measurement, and 2) the instrument mechanism, which indicates the value of the resistance being measured. There are no restraining springs attached to the indicating device in the megger *. Therefore, when the generator is not operated, the pointer floats freely, and it may come to rest at any position on the scale. Typical scales are illustrated in Figure 14-8.

The megohmmeter is expressly designed to measure very high resistances, and it derives its name from the fact that these resistances are in the "megohm" range. (The term mega means million.)

* Megger is a trademark of the James Biddle Co.

The meter, which indicates the value of the resistance being measured, is an opposed-coil type, as shown in Figure 14-9. Coils A and B are mounted on a movable member and are free to turn as a unit in a magnetic field. Coil A tends to move the pointer clockwise and coil B moves it counter-clockwise. The hand-driven generator which supplies the necessary voltage for making the resistance measurement also causes current to flow in the coils to actuate their movement. The amount of their movement depends on the resistance being measured.

When the terminals are open, current flow is from the generator through B and R_s,' which are in series with the generator. Because terminals are open-circuited, no current flows through coil A, to oppose its movement, and coil B swings counter-clockwise to a position over the gap in the core. In this position the pointer indicates infinity (∞). With the terminals shorted together, a larger current flows through one coil (A) than through the other coil (B) and the greater force in coil (A) moves the pointer clockwise to the zero position on the scale. Current limiting resistor R_1 prevents damage to the meter in this short-circuited condition.

If a resistance is connected across the terminals, current flows through coil A, R_1 and the unknown resistance R_x. This current attempts to move coil A clockwise, but the opposing force created by current through coil B tries to move it counter-clockwise. The final position of the coils is determined by the magnitude of the current through R_x, and the coils stop at a point where the forces tending to move them are in balance. The pointer then indicates the value of the unknown resistance on the scale. No springs are used in the movement because the opposing forces in coils A and B balance the pointer when a reading is being taken. Having no springs to hinder its movement, the pointer floats freely back and forth across the scale when the instrument is not in operation.

If the test leads are not connected, no current flows in coil A. However, current flows through coil B, and deflects the pointer to infinity, which indicates a resistance too large to measure. When a resistance such as R_x is connected between the test leads, current also flows in coil A, tending to move the pointer clockwise. At the same time, coil B again tends to move the pointer counter-clockwise. Therefore, the moving element, composed of both of the coils and the pointer, comes to rest at a position where the two forces are balanced. This position depends upon the value of the external resistance, which controls the relative magnitude of current in coil B. Because changes in voltage affect both coil A and coil B in the same proportion, the position of the moving system is independent of the voltage. If the test leads are short-circuited, the pointer rests at zero because the current in A is relatively large. The instrument will not be damaged under these circumstances, because the current is limited by R_1. A typical hand-driven megohmmeter is illustrated in Figure 14-10.

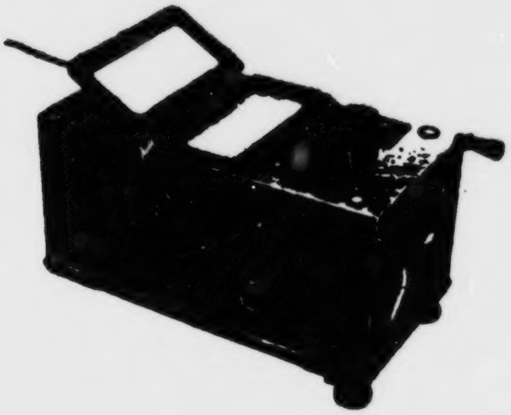


Figure 14-10 A Heavy-duty Multirange Megohmmeter

Megohmmeters can be obtained in various ranges from 500 to 2,500 volts. The higher the resistance range to be measured, the higher the voltage required to cause current flow to actuate the movement of the meter for a reading. A friction clutch restricts the generator's output to its rated voltage. The output is held to its rated value by the clutch, which is built to slip if the generator is cranked above a certain speed.

The use of a megohmmeter to test insulation resistance is non-destructive: that is, although a very high voltage exists at the terminals, the testing does not cause deterioration of the insulation.

These instruments are ideal for detecting and diagnosing faulty insulation before damage occurs. Their applications include tests on motors, wiring, lighting circuits, control equipment, switchboards, appliances, electronic equipment, meters and relays, industrial heating units, signalling equipment, aircraft wiring and equipment - nearly everything electrical.

There are other types of megohmmeters, which are used in the same manner, except that the voltage is supplied by batteries located within the instrument. This type does not require a generator, but insulation resistance can be measured in the same way. The handcrank type illustrated in

Figure 14-10 is manufactured in several sizes as required to fill the individual plant's requirements for accuracy, voltage, and compactness.

Rules to follow when making megohmmeter tests are:

1. Make sure the equipment to be tested is not energized. It should be disconnected from all other equipment and circuits.
2. Know what equipment is included in your test. The megohmmeter measures whatever resistance is between its terminals.
3. Be sure to discharge capacitors that may be charged both before and after the test particularly with large equipment.

Actually, the earth is a relatively poor conductor of electricity as compared to normal conductors like copper wire. But, if the area of a path for current is large enough, resistance can be quite low and the earth can be a good "conductor".

To understand why earth resistance or ground must be low, apply Ohm's Law to the following problem: assume that you have a 4,000 volt supply, 2,300 volts to "ground" with a resistance of 13 ohms as illustrated in Figure 14-12. Now, assume that an exposed wire in this system touches a motor frame that is connected to a grounding system having a 10 ohm resistance to ground. There will be a current of 100 amperes from the motor frame, through the fault to a ground (marked "Danger" in Figure 14-12).

If you should touch the motor frame and are grounded (perhaps by standing in a wet area), you could be subjected to 1,000 volts (10 ohms x 100 amperes). This would be more than enough to kill you instantly. If, however, the earth resistance was less than one ohm, the "shock" you'd get would be under 100 volts (1 x 100) and you would likely live to correct the fault.

Equipment can also be damaged similarly by over-voltages caused by high-resistance grounding. For your own protection, remember that only a small amount of electrical current -- as little as 0.1 ampere for one second - can be fatal! An even smaller amount can cause you to lose muscular control, and perhaps disturb the rhythm of your heart. These currents can occur in your body at voltages of 100 volts, if your skin is moist.

Use Of A Megohmmeter

In general, the higher the rated voltage of the equipment to be tested, the higher the megohm range required to measure normal as well as "safe" values of insulation resistance. Small low-voltage equipment may have very high normal insulation resistance, and, at the same time, its safe values may come well within the range of a 50- or 100-megohm instrument. The supply voltage is usually 250 or 500 volts

D-C. On high voltage electrical equipment, instruments with resistance ranges up to 2,000, 5,000, and 10,000 megohms may be used; instrument generators rated up to 1,000 or 5,000 volts are common in such meters.

The voltage should never be higher than considered safe to apply to the apparatus under test. However, the voltage rating of a megohmmeter instrument does not have to be as high as that of the equipment under test in order to obtain completely reliable readings of insulation resistance.

Five hundred volt megohmmeter instruments are most popular for general maintenance and troubleshooting. This voltage is high enough for testing many types of high-voltage equipment and yet is not considered too high to apply to ordinary low-voltage equipment.

If very large rotating machines or very long cables are to be tested, constant-voltage instruments should be used.

By using constant-voltage instruments, such as the megohmmeter, you can be confident that no higher than rated testing voltage will be applied to the apparatus under test.

The megohmmeter is one of the more common instruments you will use in your total electrical maintenance job responsibilities. Applications of this type instrument will be encountered most in equipment maintenance and troubleshooting.

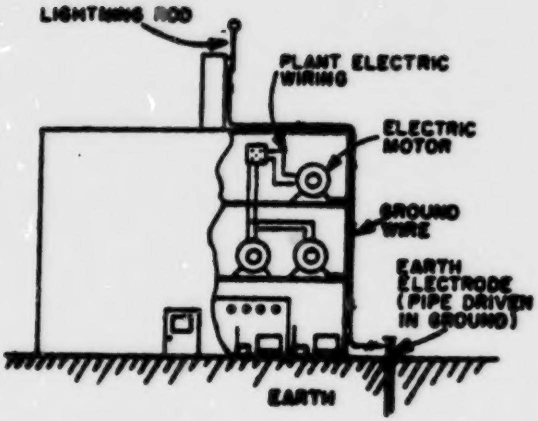


Figure 14-11 A Simple Ground System For An Industrial Building

WARNING: The megohmmeter is capable of administering severe shock voltages. Do not hold the bare contact part of the test leads while the megohmmeter is being cranked.

Understanding Insulation Resistance

Every electric wire used in an industrial plant, whether it is in a motor, generator, cable, switch, transformer, or other electrical device is carefully covered with some form of electrical insulation. The wire itself is usually copper or aluminum, which are good conductors of the electric current that powers the electrical equipment. The insulation must be just the opposite from a conductor. It should resist current and keep the current in its proper path through the conductor.

To understand insulation testing you really do not have to go into the mathematics of electricity. Ohm's Law and common sense tell us that the more voltage we have, the more current there will be. Also, the lower the resistance of the wire, the more current for the same voltage.

Good insulation has high resistance: and that resistance value can become higher or lower, depending upon such factors as the ambient temperature and the moisture content of the insulation (resistance decreases with an increase in temperature or moisture). Through the use of records, such as the resistance test record shown in Table 14-1 on the last page of this lesson, you can be made aware of the insulation condition from values that are only relative.

Proper Grounding - And Its Importance

In an industrial plant or other facility that requires a grounding system, all of the following must be carefully considered:

- Proper grounding of metallic enclosures and supporting structures.
- Protection against static electricity from friction.
- Good grounds for electric process control and communication circuits.
- Protection against induced lightning voltages.

- Protection against damage from direct lightning strikes.
- Limitation throughout the entire electrical system of any voltage to ground.

A simple and somewhat misleading conception of a "ground" for an electrical system is a section of iron pipe driven into the earth with a wire conductor connected from the pipe to the electrical circuit, as illustrated in Figure 14-11. This may not be enough of a suitable low resistance path for electric current to protect personnel and equipment.

A practical earth electrode that provides a low ground resistance is not always easy to obtain.

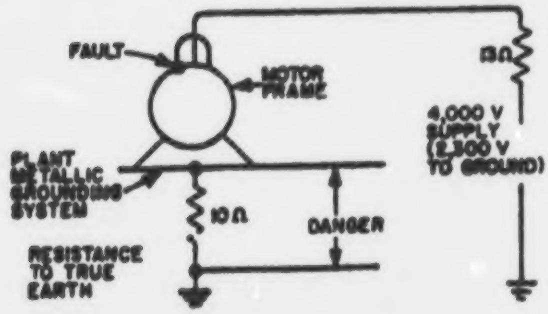
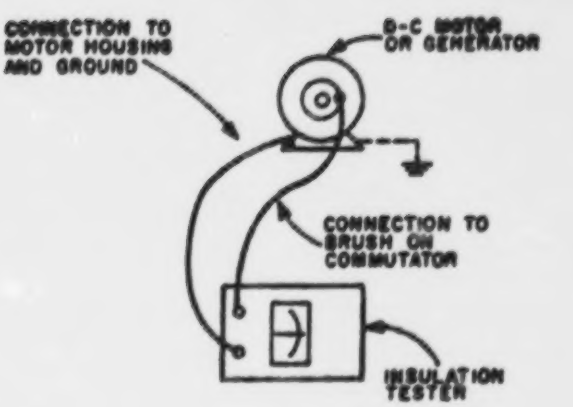
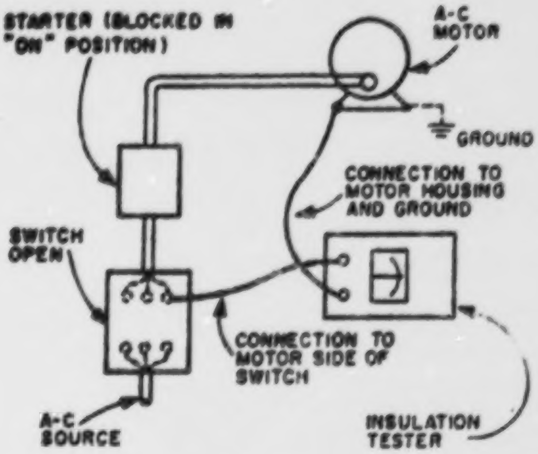

FAULT
MOTOR FRAME
13Ω
4,000 V SUPPLY (2,300 V TO GROUND)
PLANT METALLIC GROUNDING SYSTEM
10Ω
DANGER
RESISTANCE TO TRUE EARTH

Figure 14-12 An Electrical Circuit With Too High A Resistance To Ground

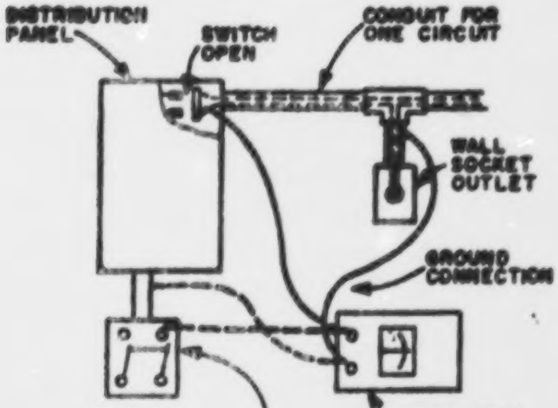
APPLICATIONS OF THE MEGOHMMETER


CONNECTION TO MOTOR HOUSING AND GROUND
D-C MOTOR OR GENERATOR
CONNECTION TO BRUSH ON COMMUTATOR
INSULATION TESTER

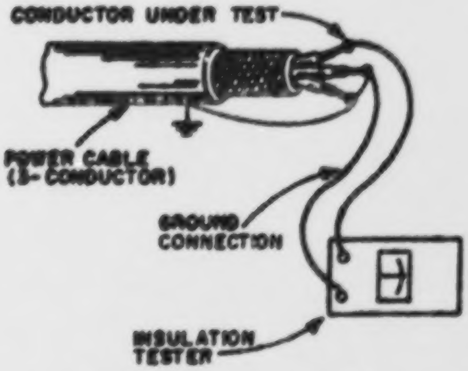
D-C motors and generators can be connected to the tester, as illustrated. This way you can test the brush rigging, the field coils, and the armature all at the same time. If an insulation weakness is indicated, you can lift the brushes off the commutator and test the brush rigging and field coils seperately. Then, with the brushes still lifted, you can connect the tester to check the armature alone.


STARTER (BLOCKED IN "ON" POSITION)
A-C MOTOR
GROUND
CONNECTION TO MOTOR HOUSING AND GROUND
SWITCH OPEN
CONNECTION TO MOTOR SIDE OF SWITCH
INSULATION TESTER
A-C SOURCE

A-C motors and starting equipment can be tested, as illustrated, by connecting one side of the tester to the motor side of the main switch; the second test connection is clipped to the motor housing. If an insulation weakness is indicated, the tester should be connected to check the motor and starter separately.

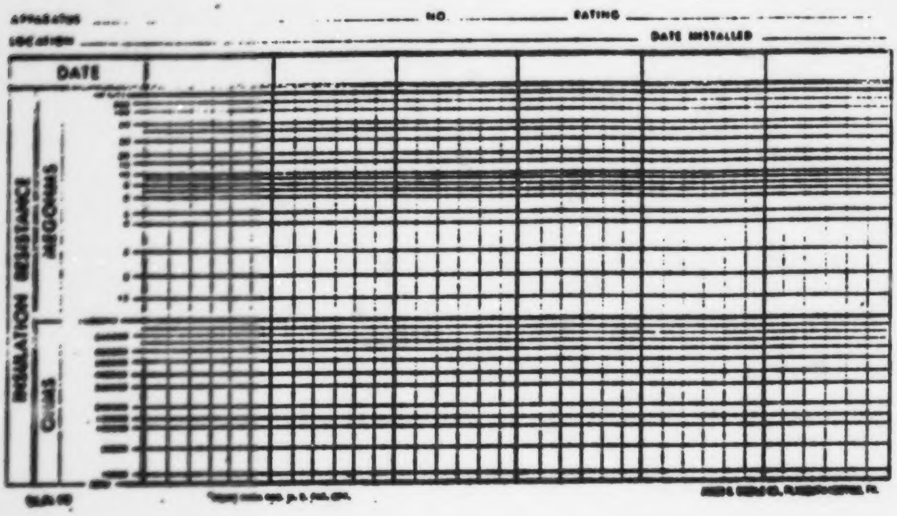


Wiring Installations can be checked from a panelboard as illustrated. The solid-line connections to the tester show how to test each circuit to ground, working from the distribution panel. Dashed-line connections at the main switch show how to test an entire wiring system to ground, provided all switches on the distribution panel are closed.



Low-Voltage Cables can be tested by first disconnecting both ends (to avoid error due to leakage across or through switchboards or panelboards), and then connecting the conductor under test to one terminal of the tester. All other conductors (two in this case) are connected together and grounded. Then they are in turn individually connected to the other terminal ground of the instrument in the same manner. Similarly, you can measure other insulation resistances such as between conductor and outside protective sheath, between conductors, and so on.

INSULATION RESISTANCE TEST RECORD



APPARATUS ________ NO. ________ RATING ________

LOCATION ________ DATE INSTALLED ________

DATE

INSULATION RESISTANCE

MEGOHMS

OHMS

INSTRUCTIONS FOR USING AN INSULATION TESTER

Checking the Instrument

Place the instrument on a firm and fairly level base. The pointer may point anywhere over the scale until the instrument is operated.

Check for an infinity reading by turning crank at a normal speed in clockwise direction. With nothing connected, the pointer should move promptly to Infinity.

Check for a Zero reading by short-circuiting the test terminals. Turn the crank slowly. The pointer should move promptly to Zero. There is no Zero adjustment.

To Make A Test

Be sure the apparatus to be tested is NOT live. Use well insulated test leads having single-conductor wire.

Check the test leads. With the leads connected to the instrument terminals and with the opposite ends separated, turn the crank at normal speed. The pointer should indicate Infinity. Touch the test leads together and turn the crank. The pointer should indicate Zero. Do not use leads that are frayed, broken or damaged.

Make the test connections and operate according to the directions on the instrument.

CAUTION: Do NOT use the instrument if its terminal or operating voltage is in excess of the safe value for the equipment under test.

ELECTRICAL MEASURING INSTRUMENTS

Electrical repairs require fast service. When production is at a standstill, it is costing the company money. Troubleshooting must be quick and efficient. It is not necessary to carry cumbersome meters to every job. Sometimes it is only necessary to have some indication that power is present.

This lesson illustrates and explains the use of simple indicating devices. The devices are instant aids, which help you determine the condition of a circuit quickly and easily. A knowledge of the use of indicating instruments will make a complicated troubleshooting task much easier to perform.

Indicating Instruments

Indicating electrical instruments are aids. They tell you whether or not voltage is present between two points. They are simple to use. They are not designed to measure the voltage, although some voltage testers indicate the approximate voltage present. A voltmeter should be used where precise measurement of voltage is needed.

The simplest indicating instrument is the light bulb. Light bulbs are used on control panels and push button stations to indicate that power is on, or in some cases, the power is off. This is a visual indicating device. For identification, coloured lenses may be used over the lamps to indicate what particular function a machine is performing at a given time.

Sometimes neon, gas-filled bulbs (or glow lamps) are used. These bulbs can be used only where the voltage is 70 volts or higher. (The firing voltage for neon is about 70 volts). The neon bulb has two electrodes but no filament. When the neon bulb is in an A C circuit, both electrodes will glow.

A neon bulb across a D C voltage will show only one electrode glowing.

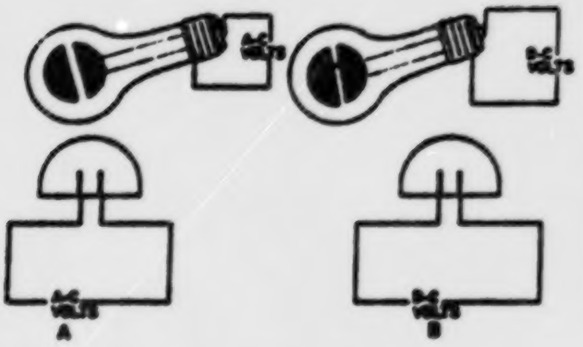


Figure 14-13 Neon Indicating Lights

The Wiggy

The **WIGGY**, or wiggins, is a trade name given to the pocket-type voltage tester shown in Figure 14-14 (A). This name comes about from the way the tester operates. A pointer moves up and down a scale and vibrates or wiggles when put across a voltage source. This indicating instrument is very handy. It can be carried in your tool pouch, available for immediate use. It eliminates carrying a delicate voltmeter around.

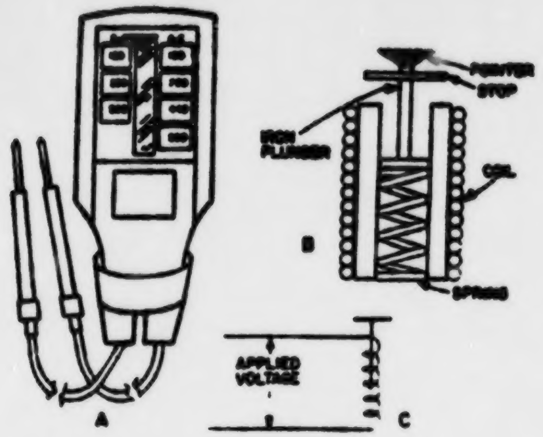


Figure 14-14 Wiggy-Voltage Tester

The construction of the wiggy is shown in Figure 14-14 (B). The principle of operation is simple. The basic component is a solenoid. When voltage is applied, current flows through the coil, which establishes a magnetic field. The magnetic field pulls the plunger downward, but plunder movement is against a spring. The spring allows the plunger to be drawn into the cylinder only a certain distance according to the voltage applied to the coil. The pointer on the plunder indicates the voltage on the scale next to it.

The voltage-indicating scale on the tester has voltage indication for A-C on one side and D-C on the opposite side. The A-C voltage scale indicates 110-220-440-550 volts. The D-C voltage scale indicates 125-250-600 volts.

Here is an example of when and how you can use the wiggy. A machine in the finishing department is down. Your foreman sends you to locate the trouble. You locate the control panel, pull the disconnect switch and open the cover. The first procedure is to determine whether power is coming in to the switch. The switch disconnects the 480 volt 3 phase supply. Three 100 amp fuses are located in the switch enclosure. Trip the safety-lock and throw the switch on. Using your wiggy, place one prod at point A as shown in Figure 14-15 (A), touch the other prod to point B. The wiggy will indicate the presence of voltage by the pointer moving down. With voltage present between points A and B, remove the test prod from B and touch point C. Again an indication on the tester should show presence of voltage. If no voltage is present, then the trouble must be at the next protective device, nearer the source which is feeding the circuit. A voltage check further down the line must be made to restore power to the switch, the same procedure should be followed. If voltage is present between the points indicated then the next step is to find the blown fuse(s).

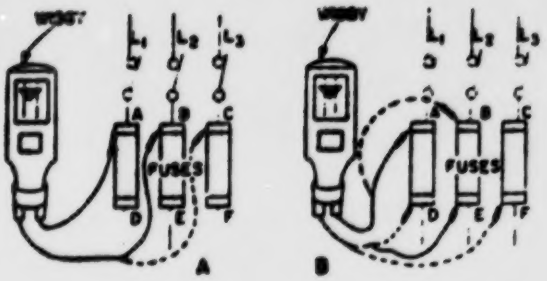


Figure 14-15 (A) Determining Presence of Incoming Power With a Wiggy.
(B) Locating Blown Fuse Using a Wiggy.

With one test prod at point A, touch point E, Figure 14-15 (B), with one other test prod for voltage between the two points. If no voltage exists, then the fuse is blown. Move test prod from point E to point F to determine if the fuse has blown. Remove the test prods and put one at point B and the other at point D. You have checked the three fuses. If no fuses are blown, then a check of the circuit downstream must be made for other possible causes of the trouble.

When any one fuse of a 3-phase system blows, it is good practice to replace all three fuses. Although one or two fuses show they have not blown, they may have heated to the point of being on the verge of blowing.

As an example - one fuse blows, and it is replaced, leaving the other two in the circuit. After checking the equipment and clearing the cause for fuse failure, the switch is thrown on and the equipment starts. Before the equipment gets running, a fuse blows, not the one that was replaced, but one of the others. Another period of downtime occurs while the fuse is replaced. Perhaps you replace all three fuses this time, even the new one you just put in. It is apparent that although nothing may be said, opinions of your performance may not be good.

Before replacing any fuses:

WARNING: PULL THE SWITCH. ALWAYS USE FUSE PULLERS to remove the fuses and to insert new fuses. NEVER USE PLIERS OR YOUR FINGERS. This is a dangerous practice.

PLIERS ARE NOT FUSE PULLERS
ALWAYS USE THE PROPER TOOLS

Continuity Testers

An ohmmeter can be used to check continuity, but this means carrying a delicate instrument with you. Pocket-type continuity testers are made commercially, but you can make one for yourself.

There are two types of continuity testers - audio and visual. The audio type tester is a simple buzzer and battery with connecting leads, as shown in Figure 14-16.

The buzzer battery combination continuity tester is used for ringing out wires. An example of its use is illustrated in Figure 14-17. Several wires of the same colour were pulled through a conduit. The wires must be identified before being connected to the circuit. At one end of the conduit, select one sire, strip the insulation so the bare wire can be attached to the conduit. At the other end of the conduit, clip one lead of the tester to the conduit. With the other lead, touch one wire at a time until the buzzer sounds. Put an identifying number (the same number) on both ends of the wire. Continue this procedure to identify the other wires.

A light bulb instead of a buzzer can be used as a visual indicator. The procedure for ringing out the wires is the same as for the buzzer.

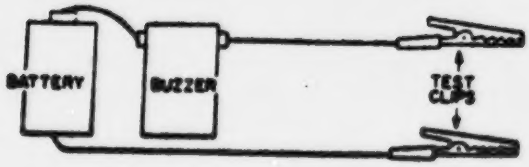


Figure 14-16 Buzzer Continuity Checker

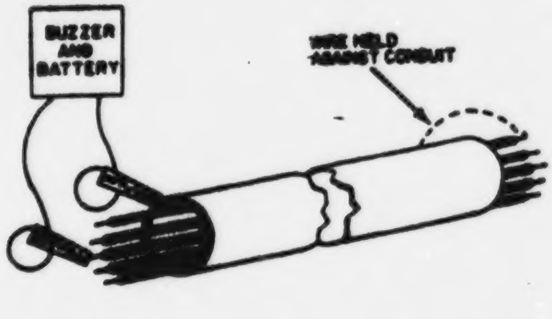


Figure 14-17 Ringing Out Wires With Buzzer Continuity Checker

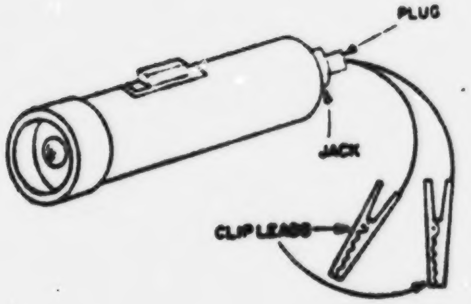


Figure 14-18 Flashlight Used as Continuity Checker

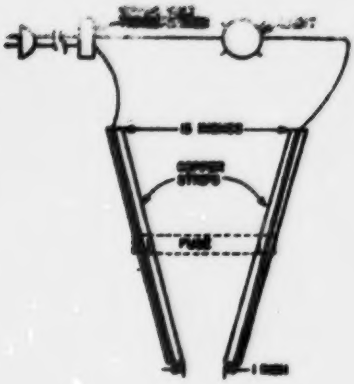


Figure 14-18 Fuse Checker

There is a flashlight available that can be used for checking continuity. The flashlight has a special cap with a jack into which a pair of test leads are plugged. When the test leads are shorted together, the light comes on. Since a flashlight is part of your tool kit, this type flashlight is ideal. It serves two purposes, and is always readily available for use. The flashlight is illustrated in Figure 14 18.

Frequency Meters

If electricity were used only for incandescent lighting, the problem of maintaining and indicating correct frequency would be of little concern. But most motors depend upon correct frequency for their rated speed. Transformers depend upon correct frequency for efficient operation. Electric clocks and timing devices depend upon frequency for their accuracy. Therefore, some type of instrument is required to indicate the frequency of the system.

A frequency meter indicates in cycles per second, the frequency of the electrical supply system. Although frequency meters are used at power plants, they can be used to monitor line frequency at any point in the system. Plant emergency or stand-by generating equipment has frequency meters so correct frequency is maintained when emergency power is required. The operator or maintenance electrician adjusts the controls of the generating equipment for proper voltage and frequency output.

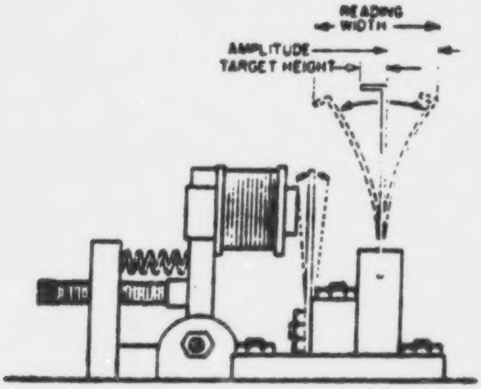


Figure 14-20 A Vibrator Type Meter and an Actuating System

Electric utility companies require frequency meters that indicate frequency to a fraction of a cycle-per-second without being affected by normal variations in the system voltage.

Since the operating frequency in this country is standard at 60 Hz frequency meters need to cover only a narrow band on either side of the normal frequency. This narrow band allows the instrument to be more accurate than it would be if it was required to measure a wide area of frequencies.

There are several types of frequency meters available. A "resonant reed" frequency meter is shown in Figure 14-20. The instrument has tuned steel reeds, which indicate alternating current frequency by resonance. The principle applied here is similar to that of a harmonica. When you blow into a harmonica, the reeds vibrate and give off certain tones. In the Frequency meter, several reeds are made to vibrate at a certain tone or frequency.

The tips of the reeds are displayed through an opening in the face of the meter. A scale is placed adjacent to the opening, and calibrated at half cycle intervals. The normal or standard frequency (60 Hz) is in the centre. This is shown in Figure 14-21.

The frequency meter described is one of the more popular types. There are other types such as the "resonant-circuit" frequency meter; another incorporates electronic circuits with direct digital readout.

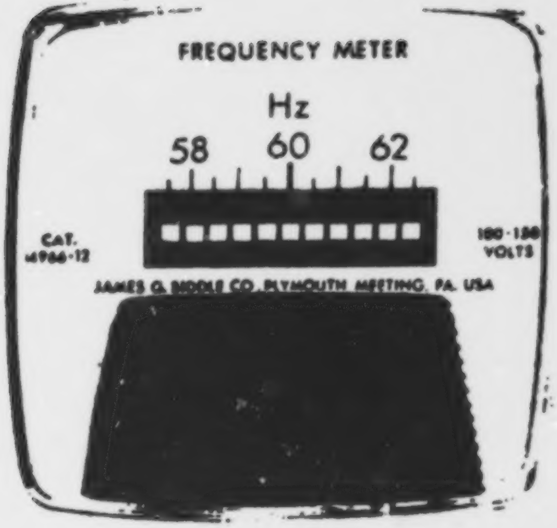


Figure 14-21   A Vibrating Reed Frequency Meter

How to Read a Frequency Meter Scale

Simply read the scale adjacent to the predominantly vibrating reed or reeds. Clear, quick reading scales permit anyone, with a little practice, to make accurate measurements without difficulty.

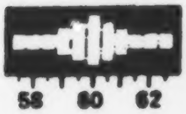
Read Directly in Hz

Reading can be easily made to one quarter of the interval between reeds providing the basic interval is on the order of 1% or less, keeping in mind our response curve is symmetrical.

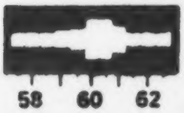
The individual reeds are guaranteed to within +/- 0.3% at 25°C in the standard instruments and to +/- 0.1% in the precision instruments.

In the examples at the right, the interval between reeds is ½ cycle. The visible pattern of vibrating reeds indicates, on the adjacent scale, the frequency being measured. With a little practice, readings can be made quickly and easily to one quarter of the interval between reeds.

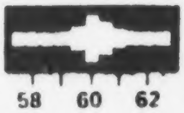
READING 60.0



READING 60.25



READING 60.125



SUBJECT: **TOPIC: 15**

PREVENTIVE MAINTENANCE **V-BELTS**

OBJECTIVES:

The trainee will be able to:

1. Identify the advantages and disadvantages of V-Belts.
2. Recall the correct method for replacement.
3. Inspection and replacing of sheaves.
4. Alignment of sheaves.
5. Safety, maintenance and trouble-shooting V-Belt drives.

V-BELT PRINCIPLES

Why V-Belts Are Used

V-Belts are reliable, relatively inexpensive, and easy to install and operate.

There has been a great increase in the use of V-Belts to power industrial drives since their invention early in this century. The advantages they provide include:

- Because they can slip, V-Belts cushion equipment from overloads and fluctuations in load.
- Alignment, although important, is not as critical as with other kinds of drives.
- They are relatively easy to select, install, and maintain.
- They operate quietly (with proper alignment and tension).
- They do not require lubrication.

....And Why Not

But they are not without disadvantages.

Disadvantages of using a V-Belt drive that may influence selection of another power transmission device, such as a synchronous drive or direct coupling, include:

- They are subject to a certain, though minimal, amount of slip, making them unsuitable for applications that require invariable, constant speed.
- They are not as efficient as other drives, although modern technology has reduced their inefficiency greatly.

How a Belt Drive Works

A belt drive transfers power from a driver, usually an electric motor, to a piece of equipment that is driven.

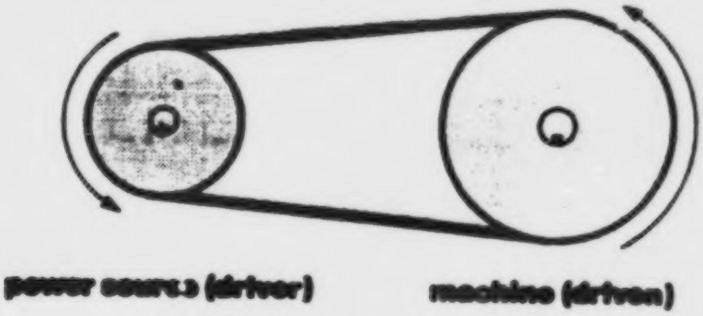


Figure 15-1 Basic Elements of a Belt Drive

Both the motor and the machinery it runs are equipped with a grooved wheel called a sheave. The belt connects the sheaves and causes the driven sheave to turn when the driver sheave turns.

A V-Belt gets its gripping power by wedging against the sides of the sheave groove.

The amount of power the driver can transfer to the driven depends on how firmly the belt can grip the sheaves. Flat belts and V-Belts both get their gripping power from tension, but a V-Belt's wedge design increases its grip on the sheave.

Increasing the tension on the belt forces it to a tighter fit in the groove and decreases the belt's tendency to slip. The right amount of tension is critical. Without enough tension, the belt will slip. With too much tension, it is likely to damage a machine's bearings.

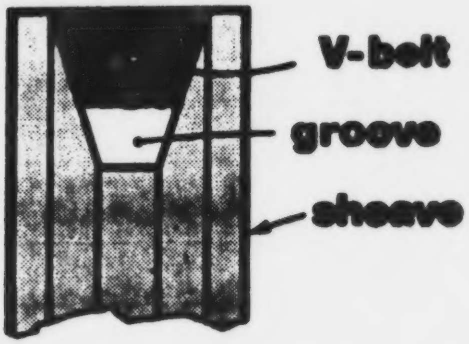


Figure 15-2 The V-Belt's Wedge Advantage

BELT WEAR

Belts, like anything else, wear out. There are several factors that contribute to belt wear, some of which can be prevented in some cases; worn or damaged sheaves, poor alignment, and belt slip.

Worn or Damaged Sheaves

Inspect sheaves frequently.

A worn, or dished out sheave reduces the belt's wedging action, and causes it to lose its gripping power. Burrs or rough spots along the sheave rim can damage a belt. When belts are changed, sheaves should be inspected for wear and damage, and they should be cleaned to remove dust, oil, and other foreign matter that can lead to pitting and rust.

Poor Alignment

Although alignment is not as critical for V-Belts as it is for other kinds of drives, running a belt on misaligned sheaves hastens belt wear.

Belt Slip

Belt slip is a common cause of belt failure and wasted power.

Belt slip is movement of the belt against the sheave, and it occurs for various reasons. Some amount of slip is unavoidable. For example belt creep, a slight forward movement of the belt, cannot be eliminated. On a well designed and properly tensioned drive, however, it is generally less than 1 percent.

Incorrect tension. The most significant cause of belt slip is incorrect tension. If slip is bad enough, it generally announces itself with a loud squealing sound. The following section, "V-Belt Replacement", describes proper procedures for setting tension on the belt to control slip.

Centrifugal force. Centrifugal force in a V-drive is the tendency for the belt to move outward from the centre of the sheave, decreasing the tension of the belt in the sheave groove. Centrifugal force is less of a problem with the narrower belts. It is more of a problem at higher speeds, with heavier belts, or with larger-diameter sheaves.

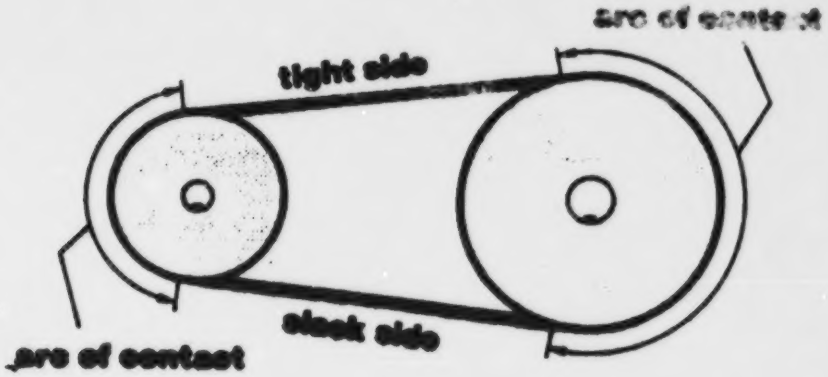


Figure 15-3 Differences in Arc of Contact

Insufficient arc of contact. The arc of contact is the amount of contact between the V-Belt and the sheave. If two sheaves have the same diameter, the arc of contact is 180° for each sheave. When the sheaves are not the same size, the arc of contact is greater than 180° on the larger sheave and less than 180° on the smaller sheave. The reduced arc of contact on one sheave results in a greater tendency for the belt to slip.

V-BELT REPLACEMENT

The first step in replacing a V-Belt is to lock out the drive.

Replacement of a V-Belt involves removing the old belt; inspecting, cleaning, and possibly replacing the sheaves; aligning the sheaves; installing the new belt; and setting the tension. But before any of these steps is begun it is vitally important to shut off and lock out the drive to prevent its operation while the belt is being replaced and adjusted.

Removing the Old Belt

Move the sheaves closer together to remove the old belt.

To remove a belt, shorten the distance between the sheaves until the belt can be slipped off easily by hand. Even if the belt has considerable slack, it should not be forced off the sheave, and it should never be pried off with a screwdriver or any other kind of tool. Such shortcuts can easily result in bruised knuckles and damaged sheaves.

Checking the Sheaves

Inspect the sheaves for damage or wear.

Sheaves should be inspected for burrs or other flaws that might damage the new belt. They can be checked by feel, preferably with the protection of a cloth or glove.

Use a sheave gauge (available from a belt manufacturer) to check the sides of the sheave groove for wear. Sheave walls should not be "dished out" or worn. An overly worn sheave will shorten a V-Belt's life considerably. Check the manufacturer's recommendations for how much wear is tolerable.

A shiny groove bottom suggests either that the sheave is so worn that the belt is bottoming out in the groove, or that the wrong size or design belt is being used. This means reduced wedging action and increased slip and belt wear. The sheave groove should be checked carefully for excessive wear.

Badly worn sheaves may also be detected in multiple-belt drives if one belt rides lower than the others.

Each sheave should be cleaned thoroughly. Dust, oil, and other contaminants can promote pitting and rust on the sheave, as well as hasten belt failure.

Replacing A Sheave

If it is necessary to replace a sheave, the new sheave must be the same type and size as the one it is replacing.

Replacement sheaves must fit the drive specifications. Submitting a sheave of a different size or type can be dangerous, particularly if a standard sheave is substituted for a precision-balanced high-speed sheave. Sheaves that are not designed for high-speed drives can shatter when subjected to the forces produced at high speeds.

Following is a list of general guidelines for removing and installing sheaves mounted with Q-D (quick-detachment) or taper-lock bushings:

1. Use the cap screws or set screws provided to back the sheave off the bushing.
2. If the bushing has been in place a long time, it may be difficult to get it off the shaft. If necessary, use a gear puller to remove the bushing.
3. Inspect the bushing and shaft to be sure there are no burrs or other flaws that could damage the new sheave.
4. If design dictates, slide the new sheave onto the shaft before the bushing is replaced.
5. Slide the bushing on the shaft, making sure the bushing is in the correct position.
6. Mount the sheave on the bushing.
7. Because of the taper, the position of the sheave will shift as the cap screws or set screws are tightened. Do not torque them down all the way because the bushing may need to be moved.
8. After the drive has been aligned, torque the cap screws or set screws to the manufacturer's specifications. The bushing screws should be and in small increments. Be careful not to over-torque.

Never use lubricants to slide the new sheave onto the shaft, and do not lubricate the bushing screws.

Alignment

The following illustrations shows ways a drive may be misaligned:

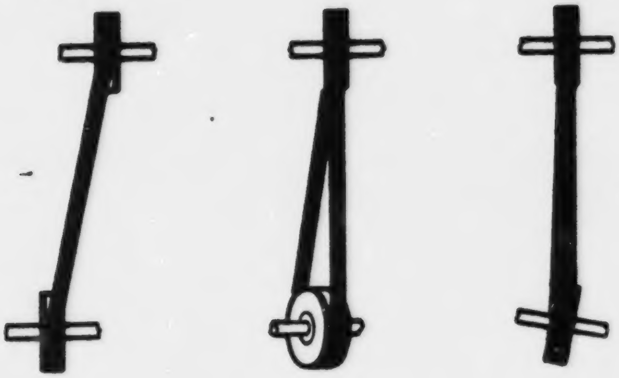


Figure 15-4 Misaligned Sheaves

Alignment of the sheaves can be checked by placing a straight-edge or a length of string across the faces of the sheaves. The string or straight-edge should run as close to the hubs of the sheaves as possible, and it should touch the sheaves at the four points indicated by the arrows.

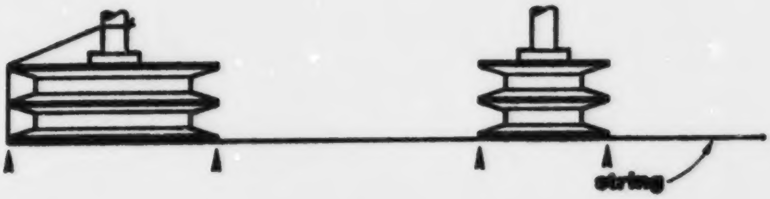


Figure 15-5 Checking Sheave Alignment

1. If the string touches both points on one sheave and neither of the points on the other sheave, the sheaves are offset and should be moved into line.
2. If it touches three of the four points, the shafts are not parallel or a sheave is warped, and the driver will have to be adjusted.
3. When the string or straight-edge touches the sheaves at all four points, move it to the other side of the hub. If the string is not touching at all four points now, adjust the driver until it does.
4. If any adjustments have to be made, check the alignment at the first position again.
5. Keep repeating this procedure, checking each side of the hub and adjusting the driver until the string touches at all four points on both sides of the hub.
6. Now rotate each sheave by hand 90° (a quarter of a turn), and check the four points again on both sides of the hub. If the string or straight-edge does not touch, the sheave is wobbly or the shaft is bent.

INSTALLING A BELT

New Belts on Older Drives

Belting requirements may need to be re-evaluated for older drives.

Conventional V-belts are significantly stronger than they were 30 years ago. They carry more load by carrying increasingly larger amounts of tension. Drives manufactured before the current RMA ratings may not require as many belts as they were designed to run. If there is any doubt about the belting requirement for a drive, work with the belt manufacturer and plant supervisor to determine how many belts to install. A few cautions should be observed:

- Some manufacturers warn that running a drive with unoccupied sheaves may lead to uneven groove wear.
- To run a drive with a reduced number of belts, the belts should be placed on the sheaves closest to the bearings.
- With a reduced number of belts, proper tensioning is even more important than it normally is.

Putting the Belt on the Sheave

A belt should never be forced into the sheave groove.

Even if the centre distance between the driven and the driver sheave has been shortened to remove the old belt, it may be necessary to shorten it a bit more to install a new, unstretched belt. The belt should slip easily over the rim of the sheave without force.

Never pry or roll a belt into the sheave groove.

Adjust the centre distance until the belt is snug.

The next step, and a very critical one, is to set the correct tension on the belt.

Tensioning

Modern drives can limit slip to less than 3 percent when the correct tension is applied.

Many people experienced with belt operation have developed a "feel" for proper tensioning. Unfortunately, "feel" that is accurate for old belts may be misleading with materials now used in belt construction. On most drives, tension can be adjusted reliably by using the force-deflection method described below. Force-deflection tensioning is not difficult, and its accuracy makes it worth the time it takes to perform.

Too much tension can result in bearing strain.

The force-deflection method is a precise way to tension a drive, but it cannot be used on all drives.

Force-deflection should not be used to tension belts on some older drives. Older drives tensioned with newer, stronger belts are likely to be over-tensioned, with the consequences of early belt failure, shaft deflection, or excessive bearing loads. Check the V-Belt manufacturer's instructions. If there is any doubt about whether force-deflection tensioning should be used on a drive, consult the belt manufacturer for the recommended way to tension that drive.

The following materials are needed to use the force-deflection method of tensioning a belt:

- Tape measure
- Spring scale
- Belt manufacturer's instructions

Force-deflection tensioning is more complicated to use with single-belt drives.

The force deflection tables described below are useful only for drives with two or more belts. To use force-deflection to tension a single belt drive, or for drives with sheave diameters and RPM ranges that are not covered in the table, contact the manufacturer for the necessary calculations.

Force-Deflection Tables

Force-deflection tensioning is based on the pounds of force required to deflect a V-Belt 1/64" per inch of span length.

—	Small Sheave Diam (inches)	Small Sheave RPM Range	Speed Ratio Range	Min Force (pounds)	Max Force (pounds)
	3.4 - 3.6	1750	2.0	3.5	5.0
A	3.8 - 4.2	to	to	3.8	5.4
	4.6 - 7.0	3600	4.0	4.6	6.6

(First Table "A")

Force-deflection tensioning, the force required to deflect the belt 1/64" per inch of span length is compared to values given in the manufacturer's force-deflection table.

Using this kind of force-deflection table requires knowing the type of belt, the diameter and speed of the smaller sheave, and the ratio of the smaller sheave to the larger sheave:

1. The first column generally indicates the belt cross-section: A, 5V, and so on.
2. Find the row in the second and third columns that corresponds to the diameter and RPM of the small sheave.

3. Because there is a direct relationship between sheave speed and sheave size, the speed ratio range (column 4) can be determined from the sizes of the sheaves. If one sheave is twice as large as the other, the smaller sheave turns twice as fast, the drive has a ration of 2.0.

4. The last two columns in the example show pounds of force that should be necessary to deflect the belt 1/64 of an inch per inch of span length.

Some manufacturers provide another kind of force-deflection table:

	Small Sheave Diam (Inches)	Small Sheave RPM Range	Drive Ratio			
			1.0	1.5	2.0	4.0 & Over
3V	2.65	1200	2.0	2.4	2.6	3.0
	3.65	to	2.8	3.6	3.8	4.2
	4.75	3600	3.8	4.2	4.4	4.8

(Second Table)

Here, each of the last four columns lists minimum deflection force for particular drive ratios. To calculate maximum force, multiply the minimum force by 1.5.

To calculate the deflection force required to tension a banded belt, multiply the tension given in the table by the number of bands.

Force-Deflection Procedure

The procedure for using force-deflection to check tension is as follows:

1. Install all belts on the drive.
2. Tighten the belts to a preliminary amount of tension-until they have a live, springy feel when they are pushed down.
3. Measure the belt span to the nearest inch. Belt span is the distance between the points where the belt first touches each sheave.
4. Use the spring scale to apply a perpendicular force to the mid-point of the span until it is deflected 1/64" for each inch of span. Use the top of the adjacent belt as a guide in measuring deflection. For example, a belt with a span of 64" should be deflected one inch; a belt with a span of 32" should be deflected half an inch (32/64).

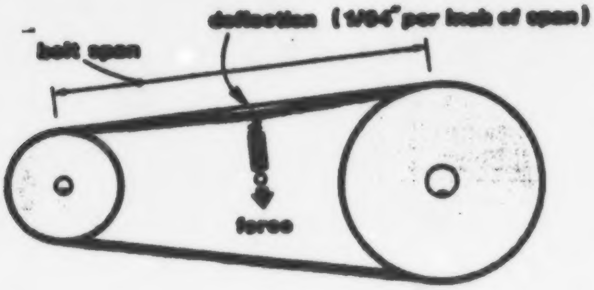


Figure 15-6 Using Deflection to Test Tension

5. Read the scale and compare the force required to deflect the belt the required amount with the minimum and maximum force recommended by the manufacturer for the particular belt and application. New belts should be tensioned close to the maximum recommendation because they will stretch.
6. If the force is less than the manufacturer's recommended minimum, there is not enough tension on the belt; if the deflection force is greater than the manufacturer's maximum recommendation, there is too much tension on the belt.
7. Either shorten or lengthen the distance between the sheaves to adjust the tension.

   If it is necessary to loosen the motor base bolts, the alignment should be checked again.

8. Belts stretch most during the first 24 hours of operation. If a belt squeal or other signs of stretch are observed, the drive should be stopped to re-tension the belts. After 24 hours, even if there are no obvious signs of belt stretch, stop the drive and re-tension the belts.

9. Stop the drive and check the tension again after 72 hours of operation, and re-tension the belts if necessary.

   Belt dressing should never be used to quiet a slipping belt.

Adjusting Tension With an Idler

Idlers can be used to adjust tension on drives with fixed sheave centres.

In situations in which neither the driver nor the driven sheave can be moved to adjust tension, an idler can be used to tension the belt. The idler shown in this illustration is an inside idler. An inside idler must be grooved to prevent belt turnover, and precise alignment is important to minimize belt wear.

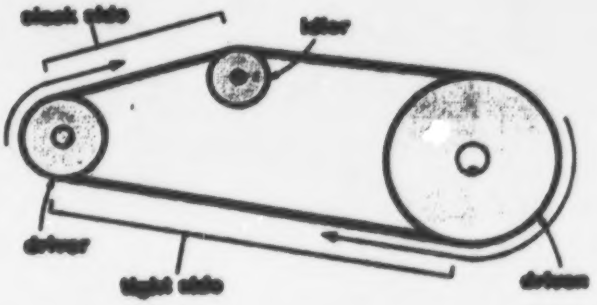


Figure 15-7 Using An Inside Idler To Adjust Tension

Outside idlers are flat pulleys that tighten the belt by pushing it inward. Although they increase the arc of contact on both sheaves, some manufacturers recommend that they not be used because they stress the belt by forcing a backward bend in it.

SAFETY, MAINTENANCE AND TROUBLESHOOTING

SAFETY

Safe practices should not be taken for granted.

Following is a list of safety reminders that should be observed for the protection of anyone in the vicinity of a belt drive:

1. Drives operating in a flammable atmosphere must be properly grounded, and belts used on them should be static conducting.

2. Any time a drive is to be worked on, it should be locked out to prevent unexpected start-up.
3. Drive guards should be used to protect people from the drive and to protect the drive from falling objects.
4. Loose clothing and jewelry should never be worn near a belt drive.
5. Hands and feet should be kept clear of an operating drive.
6. A sheave should never be operated beyond its speed capability. A sheave running faster than its design speed is likely to shatter, rocketing sheave fragments outward from the sheave centre.

MAINTENANCE

V-Belt maintenance begins with proper storage and continues with regular attention to belt tension and to the general condition of belts and sheaves.

Proper storage is the first step in ensuring maximum belt life. Belts stored in a dark, cool, fairly dry environment can be kept in storage for as long as six years with no discernable loss in service life. After six years, a 10 percent decrease in service life can be expected for each additional year the belt is stored.

After a V-Belt has been installed and tensioned according to the guidelines provided in Section 3, the following maintenance procedures will improve performance and service life:

1. Check belt tension frequently, either by observation or force-deflection as described in Section 3. Improper tension, either too little or too much, can reduce belt life by as much as 90 percent.
2. Check multiple-belt drives for uniformity of belt appearance.
3. Clean belts with a dry cloth as conditions warrant. If a belt has been splattered with grease or oil, it can be cleaned with soap and water, but flammable cleaners should not be used.
4. Belt dressing should not be used on V-Belts.
5. Check bearings periodically to be sure they are not too hot.
6. Check belt guards periodically for damage and for loose or missing bolts.
7. If a drive is not going to be operated for a prolonged period of time, the tension on the V-Belt should be decreased until preparation for start-up.

TROUBLESHOOTING

A properly designed and maintained V-drive should give a belt life of three to five years and a sheave service life of 10 to 20 years, depending on the application. Service life that falls far short of these expectations warrants investigation. The remainder of this section is a guide to preventing the recurrence of many common V-belt problems.

Excessive Heat

Belt Slip:	Check tension.
Misalignment:	Check alignment of sheaves and shafts.

Broken Belt

Shock loads or heavy starting loads:	Check tension. It may be necessary to consult with a supervisor about redesigning the drive for increased belt capacity.
Objects falling on belt or getting wedged in sheave groove:	Protect the drive with a fine-wire mesh guard.
Broken tension cords:	Probably caused by prying belt onto sheave. Follow correct procedure for belt installation.

Belt Squeal

Not enough tension:	Increase tension.
Insufficient arc of contact:	Investigate the possibility of redesigning the drive for a decreased sheave ratio.

Belt Whip

The centre distance is too long or there is a pulsating load from using too light a flywheel:	Both of these possibilities are design considerations that may require changes in the drive design.

Rhythmic Belt Vibration

Drive design problem:	Consider changing the distance between sheaves, increasing the number of belts, or using an idler sheave.

Irregular Belt Vibration

Insufficient tension:	Be sure belt is properly tensioned
Imbalanced shaft:	Check to be sure that each shaft is in balance. It may be necessary to increase the flywheel effect.
Sheaves not balanced:	Use dynamically balanced sheaves.
Framework or shafting too light:	Investigate the possibility of replacing with heavier framework or shafts.

Belt Turnover

Broken tension cords:	Probably caused by prying belt onto sheave. Follow correct procedure for belt installation.
Overload drive:	Consider re-designing the drive.
Misalignment of sheave and shaft:	Correct alignment.
Worn sheave grooves:	Replace sheaves.
Foreign matter sheave groove:	Protect the drive with a fine-wire mesh guard.
Incorrectly placed idler:	Align idler

Worn Belt Cover

Abrasive atmosphere chemical fumes:	Increase air flow around the drive.
Belt slip:	Adjust tension.

Frayed or Gouged Cover

Misalignment:	Be sure shafts and sheaves are aligned.
Damaged sheave:	Replace sheave.
Something rubbing against belt:	Clear obstruction.

Sides or Bottom Hardened or Cracked

Belt slip:	Adjust tension.
Excessive heat:	Increase ventilation.
Sheaves too small:	Consider alternative drive designs.

Belt Deterioration

Contamination from oil, grease, or corrosive chemicals:	Clean belt using nonflammable, nontoxic degreaser or detergent and water. Repair any leaks or provide additional belt shielding.
Affected by ozone:	Do not store belts near any high-voltage apparatus.

Short Worn Selections (Spin Burns)

Belt slip when starting or during periods of peak loads:	Increase tension or consider re-designing the drive.

Narrow Sections (Belt Stretch)

Broken tensile cord:	Probably caused by prying belt onto sheave. Replace belt if it is on a single drive. On a multiple-belt drive it is a production consideration whether to replace the set or leave the stretched belt until others fail.

Hot Bearings

Incorrect tension:	Check condition of belts and sheaves; adjust tension. Over-tensioning frequently causes hot bearings. In some cases, too little tension results in enough slip to cause bearings to heat.
Poor bearing condition:	Check bearing condition and lubrication.
Sheaves too far out on shaft:	Undesirable sheave placement may be forced by an obstruction; if not, reposition sheaves closer to the driver or to the driven equipment to decrease the overhanging load.

SUBJECT: **TOPIC: 16**

PREVENTIVE MAINTENANCE **BLOWERS**

OBJECTIVES:

The trainee will be able to:

1. Identify the major components of a positi displacement blower and recall the function of each.
2. Recall the installation and start-up procedures.
3. Recall the operating troubles; causes; remedies.
4. Recall likely suitable applications of blowers.
5. Recall method of determining blower performance from the published curve.
6. Recall method of determining proper impeller clearances.

BLOWERS/VACUUM PUMPS

A. GENERAL CONSIDERATION

1. Terminology
2. Pressure
3. Vacuum
4. Design

B. MAIN TYPES

1. Positive Displacement
2. Centrifugal
3. Regenerative

C. SOME DESIGN CONSIDERATIONS

1. Volume
2. Volume Control
3. Construction Features
4. Throttling
5. Lubrication
6. Materials of Construction

D. MISCELLANEOUS

1. Size of blower room
2. Sound - Silencers
3. Vibration
4. Location of Hydroelectricity
5. Piping
6. Foundations
7. Rotation
8. Flexible couplings
9. V-Belt drive
10. Maintenance
11. Repairs - Rebuilding

E. SLIDES AND OVERHEADS

F. DISCUSSIONS

BLOWERS AND VACUUM PUMPS

Positive displacement blowers have been built for over 134 years. The first one was made of wood and operated at very low pressure.

The rotary blower is a machine used to transport air and other gases from a low pressure to a higher pressure. There are two general applications for rotary blowers.

The first is pressure.

In this use, the blower usually takes air from the atmosphere and transports it to a system at a higher pressure.

The second is vacuum.

In this use, the blower pumps air from the system and discharges it to the atmosphere.

The basic characteristic of rotary blowers is they provide a relatively constant volume of air under variable pressure conditions.

These blowers provide high volumetric efficiency and incorporate many design features that contribute to job versatility and economical installation.

There are no springs or valves. This reduces maintenance and replacement costs. On the same size models, parts are interchangeable. In many cases we have been able to make one good blower out of two (2) scrap units.

The completely enclosed construction achieves compactness, provides protection against dust; and makes the unit suitable for either indoor or outdoor installations. Petrochemical plants in Montreal, Sarnia and Edmonton install all blowers and vacuum pumps outside. The reason is strictly monetary. Whenever a roof is placed over equipment, the reality taxes go up. These units operate in rain, sun, snow or sleet. Of course, totally enclosed motors are used.

POSITIVE DISPLACEMENT BLOWER

The blower is a positive displacement machine which discharges a constant volume. There is no internal compression. Pressure is not developed inside the blower, but is caused by restrictions in the discharge piping.

For wastewater treatment applications, the discharge pressure is caused by the sum of:

- pressure drop in airlines
- pressure drop through aerators
- pressure caused by the height of liquid over the aerators

A pair of figure eight impellers mounted on parallel shafts rotate in opposite directions within an elongated cylindrical housing. As each impeller lobe passes the inlet, it traps a definite volume of air and carries it around the case to the outlet where it is discharged. See Figure 1.

This cycle is repeated four (4) times during each revolution of the input shaft. The capacity of the blower is proportional to the speed of rotation, while the pressure is generated by the system resistance.

Rotation of the impellers is synchronized by a pair of timing gears which are keyed to the shafts, thereby eliminating contact between impellers.

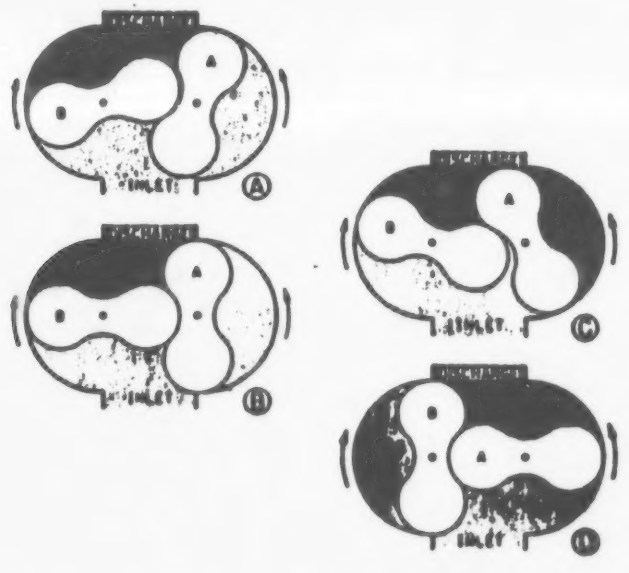


Figure 1 Compression Cycle of Straight Lobe Blower

BLOWER FEATURES

Volume

Is nearly constant over this range of pressure, having only a slight reduction caused by increase in air slip at higher pressures. At constant speed, power is approximately proportional to pressure. At constant pressure, power is directly proportional to speed.

Volume Control

Can be obtained by changing speeds, by blow-off or by-pass. Blow-off or by-pass may be accomplished with a manual or automatic valve. By-passed air or gas must be cooled before re-admitting it to the inlet.

Construction Features

Timing gears (one on each impeller shaft) control the angular position of each impeller relative to the other, and maintain the minute clearance so vital to the high volumetric efficiency of this type of machine. Since there is not internal contact or internal lubrication, a small amount of air or gas passes through these internal clearances and this quantity is termed "SLIP" (non-productive revolutions).

Clearances

A small clearance is required to permit non-lubricated operation of the impellers without having them contact each other, the cylinder, or headplates. The clearance is required to compensate for expansion due to differential pressure across the blower and growth of the impellers from the heat of compression.

Throttling

SHOULD NOT BE USED ON EITHER THE INLET OR DISCHARGE as a means of volume control. This increases the pressure rise (resulting in the horsepower required) and the temperature rise, which will eventually cause seizure of the unit due to expansion of the lobes into each other and the head plates.

Lubrication

Shaft bearings at the gear end of a blower are splash lubricated. The timing gears dip into an oil sump and distribute the oil to the gears, shaft bearings and seals. On some models the drive end bearings and seals are splash lubricated and grease lubricated on others. Those units utilizing grease lubrication on the drive end are equipped with pressure type grease fittings.

Materials of Construction

Cylinder cases, head plates, and covers are generally of cast iron. Some manufacturers utilize aluminum end covers to reduce weight. Impeller shafts are high quality steel. Timing gears, either helical or spur, are made of alloy steel.

Performance Curves

All blowers have performance curves drawn at standard conditions (barometric pressure of 14.7 PSIA, 68°F temperature and relative humidity of 0%).

To read a standard performance curve, obtain the required C.F.M. from the top left scale and follow this to the right to the point shown where it intersects with the required pressure. Blowers can make any pressure up to the maximum rating for that frame size. Since it is impossible to plot all these curves, it is necessary to interpolate between the pressures shown.

The speed at which the blower must run is read from the scale at the bottom of the curve directly under the intersecting point of pressure and CFM. The bottom right scale is used for determining the brake horsepower required to drive the blower for the pressure and volume being evaluated.

In order to determine the brake horsepower for any set of circumstances, find the intersection of the required speed and pressure from the lower portion of the curve. Read the point on the BHP scale directly to the right of this intersection.

For Example:

Using the curve for Model RVT/RB80 on the following page.

If standard conditions exist and 600 CFM of air is required at a pressure of 8 PSIG, the following procedure will obtain the required speed and brake horsepower of the blower:

1. Locate the point on the upper scale which will intersect 600 CFM on the 8 PSIG line.
2. The speed needed to produce 600 CFM at 8 PSIG is shown on the scale at the bottom of the chart directly under this intersection.
3. Locate the point on the lower scale which will intersect the required speed with the 8 PSIG line.
4. The brake horsepower needed to satisfy 600 CFM at 8 PSIG is read directly to the right of this intersecting point on the right scale.

The RVT/RB80 blower will have to run at 1840 RPM and require 29 BHP to satisfy this requirement.

LOBE-AIRE MODEL NUMBER RVT/RB80

LOBE-AIRE™

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PRESSURE PERFORMANCES
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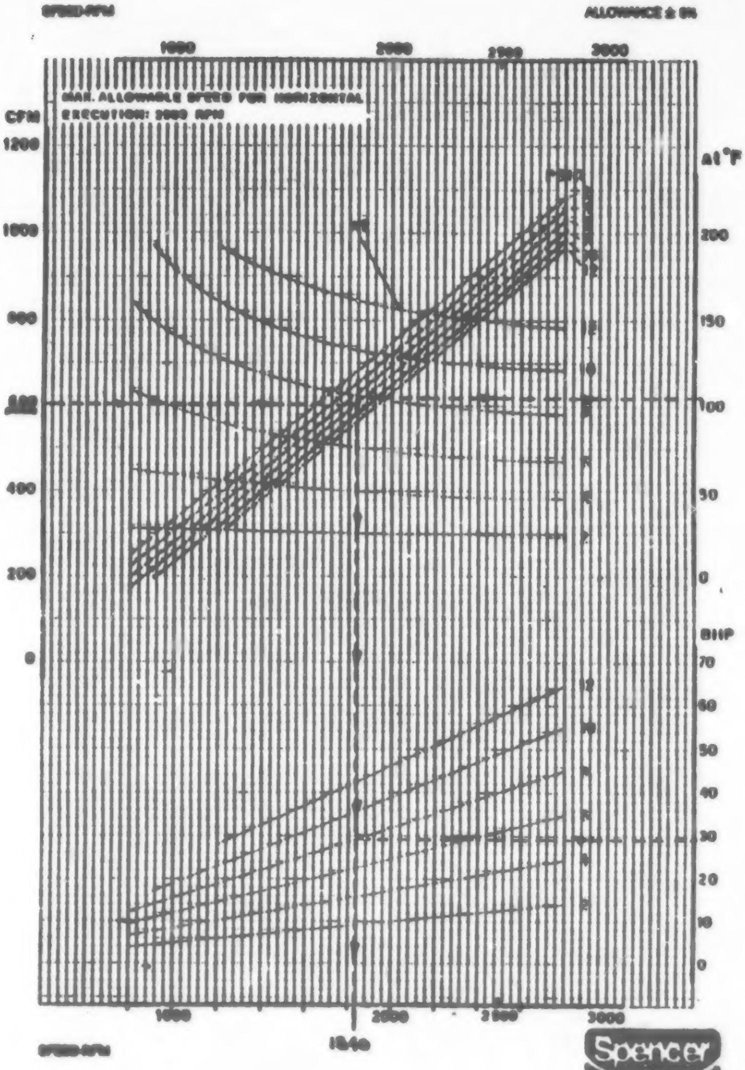
CURVE DATA BASED ON INLET CONDITIONS OF 14.7 PSIA, 68°F AND 36% R.H., SUBJECT TO A ± 5% VARIANCE

See Section 300. Engineering For Information Regarding V-Belt Drive.



JOB NAME
EQUIPMENT
CUSTOMER
CUST. P.O. #
DATE
SPENCER
PER

Spencer

Figure 2 Compression Cycle of Straight Lobe Blower

INSTALLATION

Foundations

A suitable mounting base should be prepared for the blower. Care should be taken when installing the blower that the cast iron mounting feet are not distorted when tightening the anchor bolts.

The blower must be horizontal. (This can be checked by use of an accurate machine level on the drive shaft. Some blowers have a machined surface on the top of the unit). Metal shims must be used under the feet to ensure that distortion is not transmitted to the blower cylinder.

Piping

Excessive weight of piping and fittings will cause internal misalignment and premature wear. Never allow this blower to carry the weight of the pipe. If possible, a spoor or sleeve-type expansion joint should be installed between the blower and the piping. Where a flexible connection is not practical, weight of rigid connection must be separately supported.

NOTE:

ALL SYSTEM PIPING MUST BE CLEANED INTERNALLY BEFORE CONNECTING TO THE BLOWER.

Rotation

Rotation of the motor must be checked **BEFORE** installing the blower drive. Blowers are built for reversible operation unless a rotational arrow is located above the drive shaft. If an arrow appears, the unit is timed for rotation indicated and cannot be reversed unless the unit is re-timed.

The blower should be rotated freely by hand to check for any distortion in the cylinder or foreign material that may be trapped in the blower. If the blower rotates freely, you may now install either the flexible coupling, or belts and pulleys as required.

Alignment-Drive

A. Flexible Couplings

When installing flexible couplings, care must be taken that both the drive and driven shafts are parallel and within tolerances as specified by the coupling manufacturer. This is readily ascertained by attaching a Dial Indicator to the driven (blower) shaft coupling half, and rotating the Dial Indicator around the driver shaft coupling half. Shim and move the driver accordingly. As stated, **EXTREME CAUTION** must be taken to ensure complete alignment as required by the coupling manufacturer.

B. V-Belt Drive

When a blower is belt driven, the proper selection of sheave diameters will result in the required blower speed. This flexibility can lead to operating temperature problems caused by blower speed being too low. Make sure the drive speed selected is within the allowable range for the specific blower size.

Care should be taken that the pulley faces are in line with one another; this may be established by placing a straight edge across the two front faces of the pulleys and checking the alignment. The belts should be tightened according to the manufacturer's requirements, as loose belts will cause belt failure, and belts that are too tight could cause excessive sear on the driver and the blower input bearings.

Lubrication

Ensure the oil reservoirs are filled to the proper level with the oil recommended for the operating temperatures of the unit. Most blowers have sight glasses and the correct level is the middle of the glass. If the oil falls **BELOW** the sight glass, failure of the blower due to bearing failure is inevitable. Should the oil level be **ABOVE** the centre of the sight glass, excessive heating and high oil temperatures will be experienced in the oil reservoirs and the oil will rapidly deteriorate creating the same effect as lack of lubrication.

Initial oil should be changed after two (2) weeks operation, and thereafter every six (6) months, unless the blower is running in a dirty location or subjected to high temperatures and pressures, in which case, it is recommended the oil be changed every three (3) months.

Recommended oil is a turbine quality lubricant of a single viscosity grade. "Motor oils" are not recommended, as they tend to keep any metallic parts in suspension, thereby aggravating the wear problem.

Essential additives are rust, foam and oxidation inhibitors.

OIL GRADE RECOMMENDATIONS

AMBIENT TEMPERATURE	VISCOSITY SSU @ 100°F (37.8°C)	APPROXIMATE SAE NO.
Above 90°F (32°C)	1000 1200	50
32-90°F (0 - 32°C)	700-1000	40
0-32°F (-18 - 0°C)	500- 700	30

For operation below 0°F (-18°C) the use of oil sump heaters and/or synthetic lubricants is suggested. The pour point of the oil should be at least 5-10°F below the minimum expected ambient temperature.

On some blowers which use lip type seals between the air chamber and oil reservoir, high temperature above 250°F will bake the neoprene and cause it to lose resiliency and subsequently not seal properly allowing oil to leak out.

MAINTENANCE

Preventive maintenance on positive displacement blowers will consist of regular inspection of the blower cylinder for hot spots and a regular check and replacement of the lubricating oil in the oil reservoirs as required.

The discharge pressure and temperature should be recorded regularly as a guide to the performance of the unit.

This is the extent of the normal maintenance required for this type of blower. However, should there be indication of defects internally, directions for inspection and dismantling and clearly outlined in the Operating and Maintenance Manual provided by the manufacturer.

GEAR INSPECTION

Timing gears may be inspected simply by removing the gear case after draining oil. The gear case is secured to the head plate with bolts or cap screws.

Each manufacturer usually outlines the maximum and minimum gear backlash which is tolerable for that particular size of gear.

On completion of maintenance work, be certain that gear case is restored to original position. Use a paste-type gasket compound on the mating surface or purchase a gasket from the manufacturer.

Always re-lubricate the blower before starting.

IMPELLER INSPECTION

Impellers (Lobes, Pistons), can be inspected through intake and discharge ports. This will reveal such conditions as out of time, excessive or insufficient clearance, abrasion of parts from passing foreign material, etc.

DETERMINING PROPER IMPELLER CLEARANCE

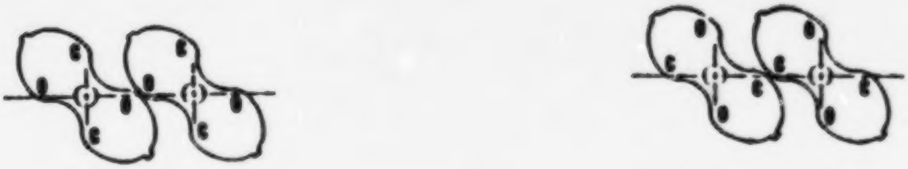
Refer to Figure 3. The clearance between impellers are measured at points 0-0 and C-C when the impellers are in the position shown. Note that the impellers are shown viewed from the drive end of the blower; always face the drive shaft end when determining clearance.

Select the illustration from the sketches which is applicable to your unit.

If drive shaft is on the right side of the centre line, it is a "Right Hand" unit - if on the left, it is a "Left Hand" unit. For vertical units: if drive shaft is on the bottom side of the centre line it is a "Bottom Hand" unit. Check above the drive shaft for a rotational arrow. If arrow appears, unit is timed for rotation only in that direction. If no arrow appears, it is "Centre Timed" for rotation in either direction.

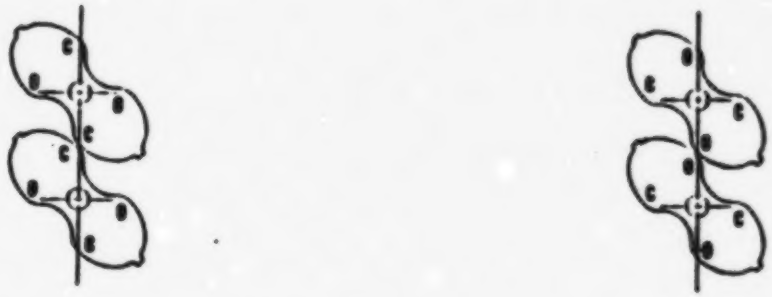
To determine total clearance, place the impellers in the 0-0 position. Measure the distance between 0-0 with a feeler gauge. Rotate the impellers 90° to position C-C and repeat measurement between C-C. Add these figures together for total clearance.

To determine correct clearances for a unit originally timed for rotation in one direction only, return the impellers to position 0-0. Divide the total clearance figure into three parts. Two parts (2/3) will be the clearance setting between points 0-0 and the remaining part (1/3) will be set between points C-C. Blowers originally timed for rotation in either direction are re-timed by dividing the total clearance evenly between points 0-0 and C-C.



Right Hand - Top Discharge
Left Hand - Bottom Discharge

Right Hand - Bottom Discharge
Left Hand - Top Discharge



Bottom Hand - Right Discharge

Bottom Hand - Left Discharge

Figure 16-3 Discharges

PRESSURE RELATIONSHIPS

Pressure is the force (weight) exerted over a surface divided by its area. Atmospheric pressure is the weight of all the air in an imaginary column extending from the surface of the earth to the edge of the atmosphere. If this imaginary column is one inch square, the atmospheric pressure is referred to as pounds per square inch absolute (PSIA). At sea level, the weight of the air in this imaginary column produces a pressure of 14.7 pounds per square inch. At an altitude of 1,000 feet (12,000 inches), there is less air in this column (12,000 cubic inches less). Therefore, the weight of the air produces only 14.2 pounds per square inch. At higher altitudes, there is less air above the surface being measured than at low altitudes with correspondingly lower atmospheric pressures.

When atmospheric air is put under additional pressure, it is measured with a gauge beginning at zero equal to the atmospheric pressure at that altitude. This is referred to as pounds per square inch gauge (PSIG). This term is frequently used to define the pressure required for an application.

On most blower applications, an inlet filter and/or silencer is required. Depending on the application, a discharge silencer may also be required.

The foregoing items, as well as piping restrictions such as elbows and tees, restrict the air and result in reducing the pressure. All these pressure factors must be considered when sizing a blower to obtain the pressure required from it. Pressure produced by the blower is measured from the inlet of the blower to the discharge of the blower. This is defined as the blower differential pressure (R P).

The schematic shows a typical installation with a pressure chart superimposed. As atmospheric air is drawn through the inlet filter and piping restrictions, the pressure is reduced. The discharge silencer and piping restrictions further reduce the pressure. In order to meet the system requirement, the blower must be able to increase the pressure for inlet and discharge losses as well as for the system pressure.

Assume an application requires 4 PSIG and atmospheric pressure is 13.8 PSIA. Pressure loss through the inlet is 0.2 PSIG and through the discharge is 0.3 PSIG. At point (A) the pressure is 13.8 PSIA. At point (B) it is 13.6 PSIA. At the blower discharge (C) pressure must be 18.1 PSIA (atmospheric pressure + system requirements + discharge losses). Blower differential is 4.5 PSI (18.1 minus 13.6). At point (D) the pressure is 17.8 PSIA. This is 4 PSI greater than atmospheric pressure and satisfies the system. The blower selection must be made on the basis of the blower differential...in this situation 4.5 PSI.

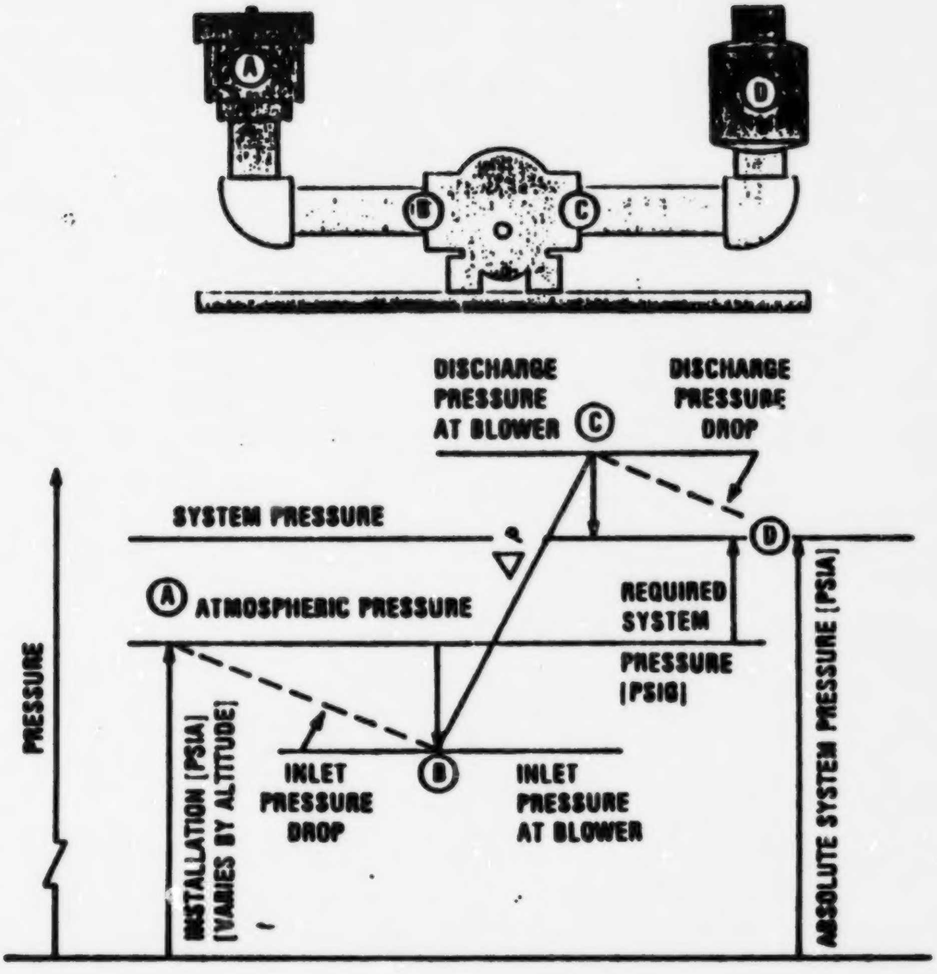

A
D
B
C
DISCHARGE PRESSURE AT BLOWER
C
DISCHARGE PRESSURE DROP
SYSTEM PRESSURE
D
A
ATMOSPHERIC PRESSURE
REQUIRED SYSTEM PRESSURE (PSIG)
PRESSURE
INLET PRESSURE DROP
B
INLET PRESSURE AT BLOWER
INSTALLATION (PSIA) (VARIES BY ALTITUDE)
ABSOLUTE SYSTEM PRESSURE (PSIA)

Figure 16.4 Schematic (not to scale)

CALCULATIONS OF SPEED (RPM)

Previously we discussed the high volumetric efficiency of the blower. Minute clearances between impellers, between impellers and end plates, and impeller tips and housing, allow for this high efficiency.

But those clearances allow air to slip from the high pressure side to the lower pressure. In doing so, air heated by compression is constantly being "recirculated".

If there is not enough new air being pulled through the blower to dissipate the heat, the temperature would escalate very rapidly.

On the Slide No. 5, the slip for various size blowers at differing discharge pressures and vacuums is tabulated.

Note the slip (non productive revolutions) increases directly with pressure.

SLIP TABLE - RB SERIES

PRESSURE SLIP TABLE

Unit Size	Pressure - PSIG														
	1	2	3	4	5	6	7	8	9	10	11	12	13	14	15
70	142	212	257	297	337	372	[illegible]	417	437	467	481	502	534	567	[illegible]
80	[illegible]	198	238	270	[illegible]	[illegible]	[illegible]	398	418	438	463	[illegible]			
81	140	217	263	307	336	372	384	427							
90	112	181	201	229	251	279	301	329	346	360	382	399	414	434	[illegible]
100	106	157	185	220	246	[illegible]	[illegible]	321	334	352	363	379			
101	[illegible]	173	[illegible]	230	[illegible]	[illegible]	311	341							

Let us do the mathematical calculation for a blower having a capacity of 600 SCFM @ 8 PSIG.

From reading the performance curve previously, we selected an RVT/RB80.

An RVT/RB80 blower has a displacement of 0.416 cubic feet per revolution (CFR).

The number of revolutions required is determined by dividing the required capacity by the CFR.

Therefore, Productive speed = 600 ÷ 0.416 = 1442 RPM

At 8 PSIG the "Slip" expressed in revolutions per minute is determined to be 399 RPM.

To get the required speed the "Slip" must be added to the productive speed.

Actual speed = 1442 + 399 = 1840 RPM

As you can see "Slip" can be 21.6% of the actual RPM, which is a significant amount of lost output.

Volumetric efficiency of blowers is generally in the 75% to 83% range.

When foreign material passes through a blower, wear takes place, and the clearances open up, allowing more air to slip from higher pressure to lower pressure.

The slip increases and the efficiency of the blower is reduced.

At the same time, the discharge temperature starts to rise because more heated air is slipping back to the suction side.

HORSEPOWER CALCULATIONS

Because of the design of the positive displacement blower, horsepower required is strictly a function of speed and discharge pressure.

B.H.P. = RPM x CFR x P (PSIG) x .00474 (constant)

Using our previous examples of an RVT/RB80 producing 600 SCFM @ 8 PSIG, calculated horsepower will be as follows:

B.H.P. = 1840 x 0.416 x 8 x .00474 (a constant) = 29.025

Standard conditions and requirements for the RVT/RB80 blower are as follows:

CAPACITY	600 SCFM
DISCHARGE PRESSURE	8 PSIG
DISPLACEMENT	0.416 CFR
SLIP	399 RPM
ACTUAL SPEED	1840 RPM
BHP	29

In the event of foreign material passing through the blower and increasing clearances, the "Slip" will increase, resulting in a lower output; but the horsepower requirement will remain the same, for a fixed speed.

This is the main reason to ensure the air filters are kept in good condition.

TEMPERATURE RISE CALCULATION

The formula for temperature rise through a blower is as follows:

TEMPERATURE RISE $= \dfrac{T_i \times \text{BHP} \times 0.92}{.00436 \left(\dfrac{K}{K-1}\right) \times \text{PSIA}_i \times V_i}$

T_i = Inlet Ambient Temp °F + 460°R

BHP = Horsepower Required

0.92 = A Constant

$$K = \frac{C_p \text{ (Specific heat - Constant pressure)}}{C_v \text{ (Specific heat - Constant volume)}} \quad \begin{array}{l} = 1.395 \text{ for Air} \\ = 1.316 \text{ for Methane} \end{array}$$

PSIA inlet = Inlet Pressure on the Gauge + 14.7 = Absolute Pressure

V inlet = Inlet Volume in cubic feet per minute (CFM)

For Air at Standard Conditions

$$.00436 \left(\frac{K}{K-1}\right) = .01542$$

In our example of 600 SCFM @ 8 PSIG we have assumed standard conditions of 70°F and an elevation of sea level, which is 0 PSIG on a pressure gauge.

Temperature rise $= \dfrac{(70 + 460) \times 29 \times 0.92}{.01542 \times (0+14.7) \times 600} = 104°\text{F}$

Discharge temperature = 104 + 70 = 174°F

At standard conditions of 70°F and sea level, this equation can be simplified to:

Temperature rise $= 2150 \times \dfrac{\text{BHP}}{\text{V Inlet}}$

Don't forget to add the 70°F to get discharge temperatures. If mathematics is not your strong point, just remember:

For every one (1) pound of pressure rise,
the discharge temperature will be increased
by about 13°F above ambient.

FOR EXAMPLE:

With a blower discharging at 8 PSIG discharge temperature should be about:

8 x 13 + 70 = 174°F

COMPARISONS OF ROTARY POSITIVE vs CENTRIFUGAL BLOWERS

(A) NOISE

Centrifugals are generally considered to be quieter operating. At a normal speed of 3500 RPM, the frequency of noise is higher and these higher frequencies are more easily absorbed and muffled.

Rotary positive blowers have two (2) main sources of noise: (1) mechanical (gears); and (2) air pulsation. These are of much lower frequency and more difficult to muffle, hence more attention, design and equipment are required.

(B) VARIABLE VOLUME

Air requirements (oxygen requirements) for a given size plant may change due to two (02) main factors:

1. Substantial increase in the volume of sewage entering the plant - since aeration tanks have a fixed volumetric capacity, a substantial increase in flow lowers the retention time in the aeration tank, therefore, more air must be supplied to meet the oxygen demand.

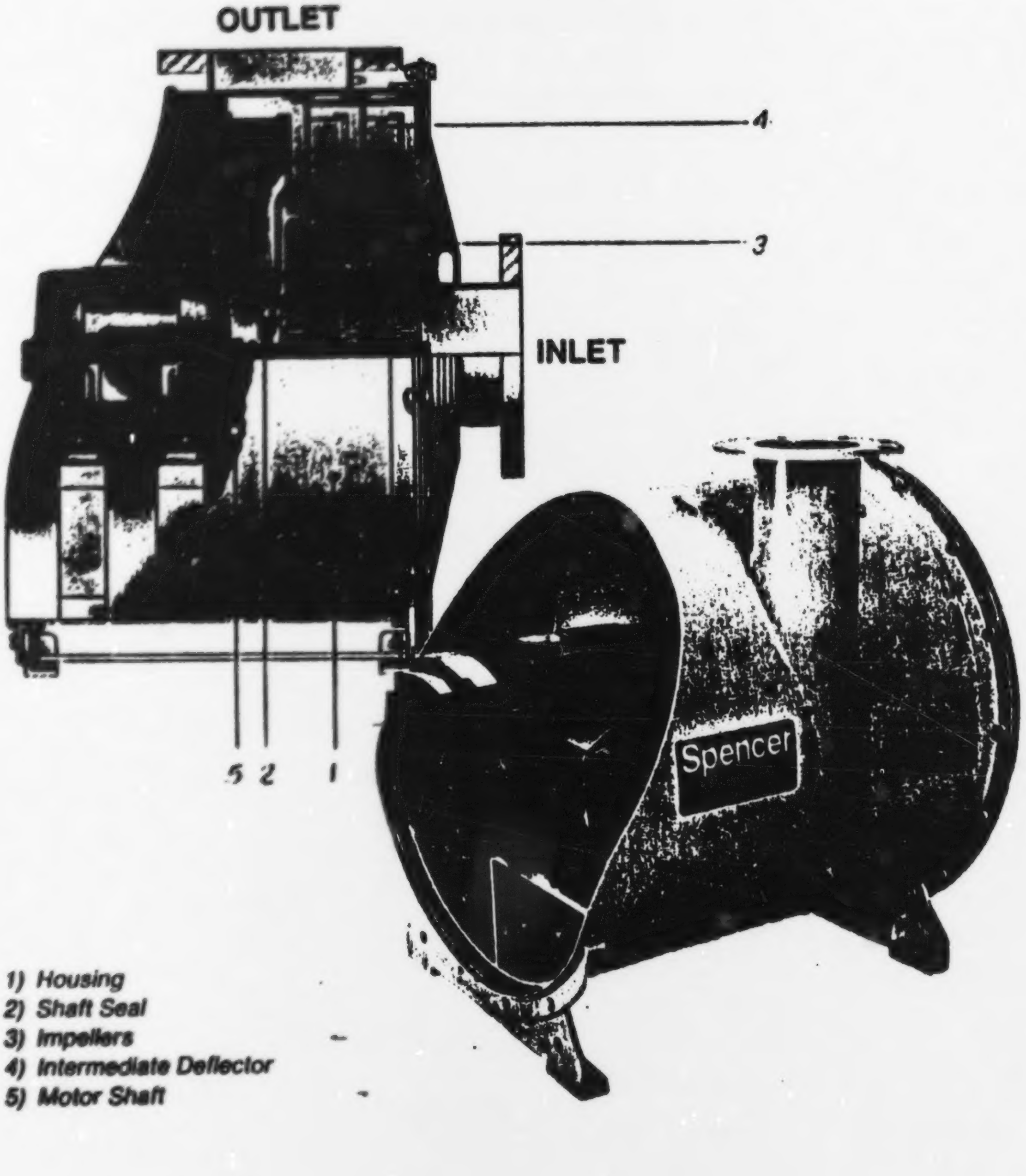
   Sewage volume increases can be:

   (a) at certain periods of each day (percentage of increase is usually small)

   (b) seasonal (i.e. in resort areas or near (large food canning facilities)

   (c) as a result of increased building of residences or factories being served by Pollution Control Plants. This is a long term, gradual increase, usually several years. (Most Pollution Control Plants are built and designed for future expansion)

CENTRIFUGAL BLOWER



1) *Housing*
2) *Shaft Seal*
3) *Impellers*
4) *Intermediate Deflector*
5) *Motor Shaft*

2. Marked increases in sewage strength is the second reason air requirements change. Increases in strength are usually due to periodic discharges by local industries into sewer lines. This is usually on a scheduled basis it may be daily or weekly.

RECOMMENDATIONS

Volume variations could be handled as follows:

1. **Daily**

   Select blower to provide air for maximum requirements since this is how most centrifugals are run, or size unit for normal load and have a standby unit run only during time of peak loads. It could easily be started automatically through a clock timer.

2. **Seasonal**

   Provide an extra set of sheaves which will meet higher capacity during peak load season. Gives maximum power savings.

3. **Future Capacity**

   Select a blower to suit present design requirements. As plant expands add new blowers as required. Most plants are built and expended in increments. It does not pay to provide large centrifugals and run them for several years at reduced capacity. This is inefficient operation and poor investment of money. It is important to provide standby capacity. This is easier to do with smaller units.

MAINTENANCE

This centrifugal blower has one (1) rotating assembly (impellers and shaft) supported by two (2) bearings. The rotary positive has two (2) rotating elements held in place with timing gears. Since the centrifugal has greater clearances and fewer rotating parts, it is generally assumed to be simpler and less expensive to maintain.

EFFICIENCY

At standard inlet conditions (14.7 PSIA, 70°F) the centrifugal uses 15% or more horsepower than a rotary positive blower.

CAPACITY (PRESSURE AND VOLUME)

a) Pressure output of a centrifugal is very much affected by altitude and/or inlet temperature.

Pressure capability of a rotary positive unit is not affected by altitude or temperature.

Result - Guaranteed pressure under all conditions.

(b) Inlet volume of a centrifugal is also affected by altitude and/or temperature, up to a 22% variation.

Volume output of a rotary positive blower is also affected by altitude and temperature but to a much lesser degree, usually within a couple of percent throughout the whole temperature range.

DESIGN AND PERFORMANCE

COMPARISONS

CATEGORY	CENTRIFUGAL	POSITIVE DISPLACEMENT
RUNNING CLEARANCE	RELATIVELY LARGE	RELATIVELY SMALL
PRESSURE DEVELOPMENT	VELOCITY CHANGES	MECHANICAL ACTION
FLOW CHARACTERISTICS	VARIABLE VOLUME/CONSTANT PRESSURE	CONSTANT VOLUME/VARIABLE PRESSURE
TEMPERATURE CONSIDERATIONS	RELATIVELY INSENSITIVE	RELATIVELY SENSITIVE
NOISE LEVELS	RELATIVELY LOW	RELATIVELY HIGH
MATERIAL INGENSTION	RELATIVELY TOLERANT	RELATIVELY INTOLERANT

HORSEPOWER

Besides being less efficient at standard conditions, the horsepower required by centrifugal varies inversely as the temperature and altitude.

$$\text{BHP at site} = \text{BHP at Standard Conditions}$$

$$= \times \left(\frac{\text{PSIA at SITE}}{14.7}\right) \times \left(\frac{530}{\text{Abs. Temp. at Site}}\right)$$

Obviously BHP required is related to actual work being done. Since less work (lower pressure and volume) is being done the horsepower is reduced from that required at standard conditions.

However, what happens when temperature drops to minus 10°F? More pressure is developed, more volume is handled, and more horsepower is required.

Under conditions above, if temperature drops to -10°F, the BHP required at site rises 18%!

The consultant must be very careful to spell out the extreme high and low temperatures, as well as volume in SCFM, when sizing centrifugals.

Rotary positive horsepower requirements are based on blower speed, pressure differential, and CFR (displacement). These are constant throughout the temperature range.

ENGINE DRIVE CAPABILITY

When digester gas is available for driving the aeration blowers, rotary positives are almost always chosen, because they are much more adaptable to the slow engine speeds and varying speeds engines can provide.

Occasionally on very, very large plants, slow speed centrifugals may be used.

With engine drives, volume output on rotary positives can be varied.

EQUIPMENT COSTS

In almost every case where bids have been requested on both centrifugal and rotary blowers, the costs of a positive displacement blower with all necessary accessories has been lower.

PARALLEL OPERATIONS

When two or more centrifugals will be running simultaneously and discharging into a common header, identical units (same volume and pressure ratings) will be recommended. This is necessary at the pressures required in Pollution Control Plants to reduce "hunting" or "surging" between units. Surge is a pulsating thrust which is hard on the bearings.

With rotary positive units, blowers with different volume ratings can be utilized. This is an advantage in large plants where air requirements for normal loads, peak loads, near future additional loads and standby must be considered.

Rotary positives have flexibility in sizing.

G. OPERATING TROUBLES - CAUSES - REMEDIES

CAUSES	REMEDIES
1. EXCESSIVE DISCHARGE PRESSURE	
Air capacity over the necessary amount of the plant	Reduce the rotational speed of the blower, when possible; blow off in the open air or by pass part of the gas (refer to Chapter B)
Obstructions or throttlings into discharge pipe	Remove the causes of obstructions or throttlings
2. SAFETY VALVE CONTINUOUS OUTFLOW	
Excessive discharge pressure	Reduce the pressure to the allowed value (see point 1)
Valve setting too low in comparison with the plant pressure	Set again the valve, increasing the breathing pressure to the allowable value
3. AIR OR MACHINE OVERHEATING	
(It is normal the air heating of 10°C, over ambient temperature for each 1000 mm W.C. (100 mbar) back pressure)	
Plant's pressure higher than treated pressure	Increase discharge pipe diameter; remove possible obstructions or throttling; reduce when possible blower rotation speed release in the open air part of compressed air
Clogged inlet air filter	Clean the filter or replace the cartridge
Too viscous lubricant oil too low oil level	Replace the oil with another less viscous; settle the right level again

CAUSES	REMEDIES
Excessive wear of rotors and consequent increase of clearances	Check the clearances; consult the manufacturer
High room temperature due to scanty ventilation	Fan the room

4. ANOMALOUS VIBRATIONS AND NOISES

Excessive wear of rolling bearings	Check the rolling bearings; replacement is necessary when the clearance is over the value C_a
The rotors meet owing to out of phase as consequence of overloads or foreign bodies entry	Check the back pressure; disassembly and replace the phase; before assembling check the rotors are perfectly aligned
Rotors mutual sliding or rotor casing sliding due to foreign materials or dust	Clean the blowers; check that the covers of the casing have not been damaged
Rotor casing covers sliding due to overheating	Verify the air filter; check back pressure; disassemble and check condition of the covers
Rotors out of balance due to deposits or foulings	Clean rotors and casings: check the rotors are aligned

CAUSES	REMEDIES
5. LOW AIR CAPACITY	
Clogged inlet air filter	Clean the filter
Excessive suction depression	Settle the stated value of the suction pressure again
Belts drive slip	Stretch the belts
Wear of the rotors with too high increase of clearances	Measures the clearances; consult the manufacturer
Gas leakages from delivery pipe	Check the delivery pipe
6. MOTOR OVERLOADING	
Differential pressure, back pressure or suction pressure too high in comparison with rated values	Act on the plant to reduce pressure to rated values
Blowers capacity too high in comparison to requested plants capacity: consequently pressure increase	Let air surplus leak in the open air or better reduce rotation speed of the blower
Clogged inlet air filter	Clean the filter
Rotating parts meeting and internal slidings	Stop the blower immediately (see point 4)

APPLICATION GUIDE
BY INDUSTRY

The following list is only a sampling of applications in which blowers have been used:

ABRASIVES	pneumatic conveying; vacuum pick up of sand for blasting
AIRCRAFT	instrument and ground control cooling; air leakage test supply for cabin coolers; carburettor testing; engine overhaul check; ground testing of pneumatic auxiliaries; air agitation of plating tanks, cleaning and pickling tanks; air leakage testing of fuselages; altitude chamber testing of instruments
ALUMINUM	pneumatic conveying of bauxite powder from barge to storage; separating aluminum and sand particles by flotation on air
AQUARIUMS	aeration
ASPHALT REFINING	oxidation by aeration
AUTO WASHING	air blasting for drying
BAGS	vacuum folding of paper bags; opening folded bags; vacuum pick up of flaps on envelopes for filling; air supply for bulk material in bag filling
BAKERIES	pneumatic conveying in plant and from trucks to storage bins
BEARING MANUFACTURE	conveying steel chips; heat treating furnaces
BREWERIES	conveying malt and grain; bottle and can drying; suction pick up of bottles and cans for packaging; pumping CO_2 for carbonation; vacuum drying of grain
BUILDING, INFLATABLE	inflating synthetic rubber buildings for warehouses, shelters, etc.

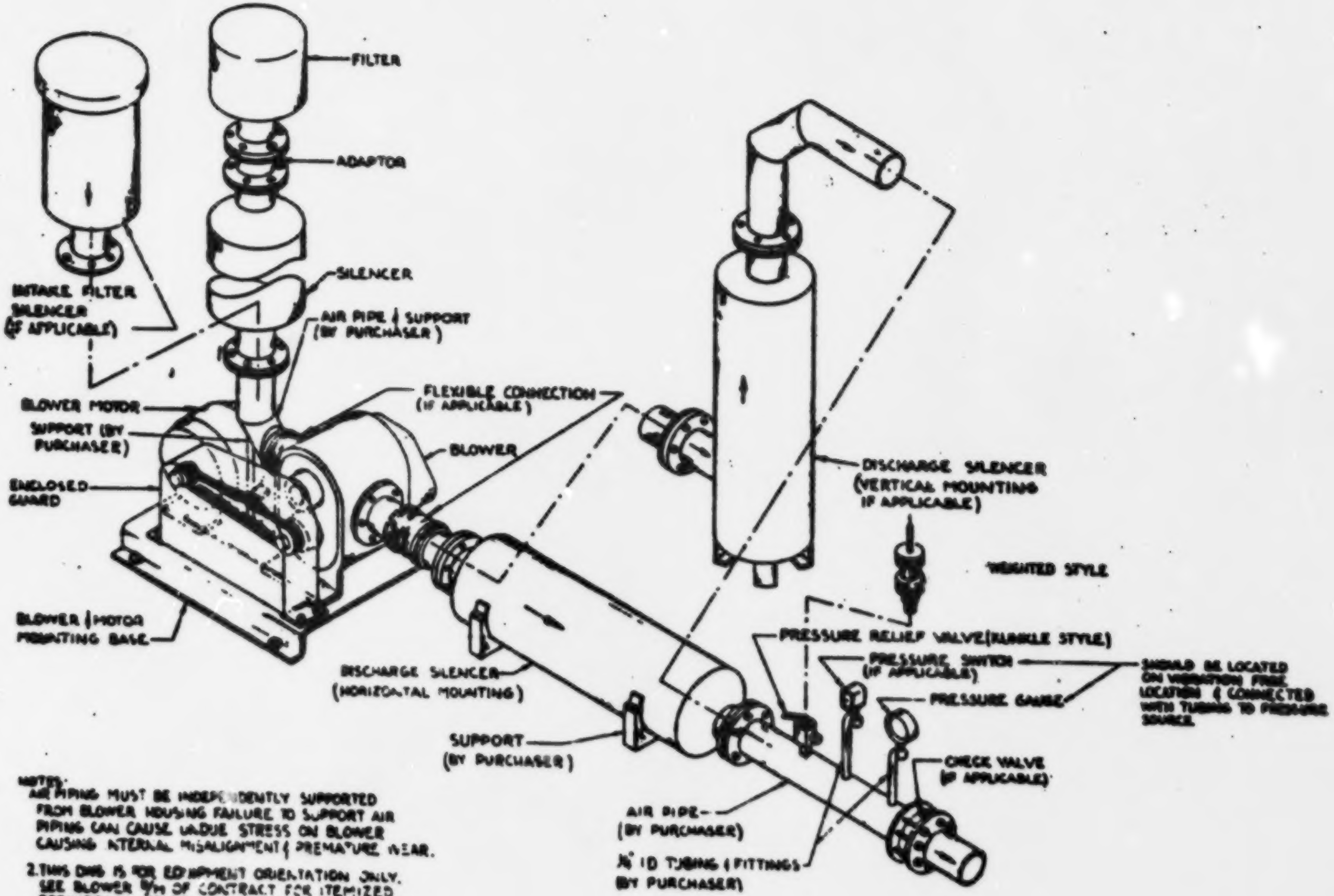
CANDY	pneumatic conveying of sugar, flour, and other ingredients
CARBON DIOXIDE	pumping CO_2 gas for making dry ice
CARBON BLACK	pneumatic conveying and bagging; combustion air
CEMENT	air supply to homogenize bulk cement going into storage bins; fluidizing cement for conveying in batch plant; conveying from truck to storage bins; air slide for bulk cement transport trucks; conveying ready-mixed concrete batch plants to trucks
CLOTHING	inflating fabric bag form for shirt pressing; vacuum drying
COAL	rock dusting in shafts, tunnel roofs and walls; air sampling; coal washing; tunnel ventilation; vacuum priming water pumps; dust collection from roof bolt drilling; vacuum drying of coal slurry; coke blowing on vacuum filters; vacuum filtration for refuse from coal washers
CONTAINERS	vacuum pick up of bottles and cans
CONVEYING	pneumatic, vacuum, or combinations for materials such as: sands cement coal fly ash bauxite grains starch feeds sales cereal gypsum flour sugar carbon black transfer materials from railroad cars, trucks ships to vats, hoppers, bins
CREOSOTING PLANTS	air evacuation in vacuum chambers
CUPS	conveying products through tubes to packaging point.

CURTAINS, AIR	furnaces; heat treating
CUSHIONS	air cushions or glides by blowing air to facilitate easy movement of flat pieces
DISTILLERIES	liquor moving, pneumatic conveying of grains; emptying aging barrels
DROP FORGING	scale blowing from forges
DUST COLLECTION	cleaning bag filters; bag filtration
ENGINES	air scavenging; supercharging; air testing
FARMING	drying; conveying; spraying; soil pasteurization
FISH	aeration of fish tanks, transport trucks, and ponds; prevention of ice formation on ponds; removal of eggs from fish; agitation of washing tanks
FLOUR	fluidizing for conveying
FOOD PROCESSING	freeze drying; evacuation of air from bottle or can filling machine; agitation; pneumatic conveying
GAS MIXING	proportioning gases in chemical processes
GAS & OIL FIELDS	field repressurization from exhaust gases of gas engines
GLASS	pneumatic conveying; vacuum holding of sheet glass for transfer; evacuation of test changer; vacuum filtration from grinding and cutting particles; air cooling sheets and moulds; blowing water from glass sheets, cooling and tempering air for curved windshields leaving moulds
GRAIN	pneumatic conveying; air drying; air circulation; supplying air for fermentation

GYPSUM BOARD	pneumatic conveying of bulk gypsum; vacuum for board machine; vacuum lifting of boards for transfer
ICE	aeration of lakes and ponds for protection against freezing; agitation by bubbling water in marinas, dams, bridges, etc., where freezing may cause damage; aeration of harbours to keep open for navigation; aeration of freezing tanks in ice plants to draw off impurities and to provide clearer cakes of ice; conveying crushed ice into ice bunkers on boats, ships, trucks and railroad cars
INSULATION	blowing insulation into walls
IRON ORE PROCESSING	vacuum filtering to dry iron ore for pelletizing; cake blowing; combustion air for pelletizing furnaces and for drying wet ore; pneumatic conveying; flotation air; cooling pellets; cooling bearings on pelletizing furnaces
LAUNDRY	shirt pressing machines; vacuum drying of clothing
LEAD	combustion air for lead melting furnace
MACHINERY MANUFACTURING	vacuum for holding material in place for surface planning and grinding; vacuum on chucks for holding material in place; dust collection
NON-FERROUS METALS	aeration of slurry
OIL REFINERIES	asphalt blowing; fly ash conveying; liquid transfer by air pressure; vacuum flashing; pneumatic conveying
ORE PROCESSING NON-FERROUS	copper converters; flotation for separatting copper sulphides from waste; vacuum filtration; cake blowing; combustion air
PACKING PLANTS	pneumatic conveying; agitation

PAPER MILLS	paper machine vacuum; head box pressurization; pneumatic conveying; waste treatment aeration; paper folding; vacuum pick-up of sheets; air glide or cushion for moving heavy loads on flat surfaces; vacuum cleaning/conditioning of felts; roll heading for wrapping; black liquor recovery
PHOTO PROCESSING	holding photos for drying; blowing water off set film
PLASTICS	blowing granules into moulding machine; inert gas conveying to prevent contamination of polyurethane; hopper drying of material going to machines; hopper filling; pneumatic conveying
PLATING	agitation; moisture blowing and drying
POULTRY	vacuum removal of entrails; agitating water in washing tanks; removal of egg from shells; pick-up of eggs for packing in cartons
PUMPS, CENTRIFUGALS	vacuum priming to lift water into pumps
PRINTING	paper folding machine; control air glide between rolls; collating; mat drying; vacuum pick up of sheets
PULP MILLS	chip conveying, combustion air; vacuum filtration; pulp drying; black liquor recovery
ROAD CONSTRUCTION	asphalt paver combustion air; convey cement slurry under sagging pavements, etc.
SEPTIC TANKS	vacuum cleanout of tanks; pressurizing truck tanks at disposal area
SEWAGE	aeration in activated sludge plants - industrial, municipal, commercial (hotels/motels s/sub-divisions/ships, etc.); separation of grit from sewage; grit channel aeration (prevents settling of grit); pneumatic conveying; dewatering sludge on vacuum filters; recirculation of sludge gas; cake blowing; combustion air for incineration

STEEL MILLS.	blast furnace air supply; combustion air for open hearth furnace; bessemer converter; cupola blowers; coke oven exhauster; fly ash collectors; vacuum filtration, agitation of ferrous sulphate
SUGAR	pneumatic conveying; aeration of storage bins; vacuum filtration
TEXTILES	vacuum for drawing fluid through bolts of cloth; pneumatic conveying; drying yarn after dyeing
TUBE SYSTEMS	pneumatic conveying
WATER TREATMENT	vapour compression; water distillation; aeration of streams; lime neutralization aeration; air flocculation for water purification; saline water de salinization; salt water de salinization


FILTER
ADAPTOR
SILENCER
INTAKE FILTER SILENCER (IF APPLICABLE)
AIR PIPE & SUPPORT (BY PURCHASER)
FLEXIBLE CONNECTION (IF APPLICABLE)
BLOWER MOTOR
SUPPORT (BY PURCHASER)
BLOWER
ENCLOSED GUARD
DISCHARGE SILENCER (VERTICAL MOUNTING IF APPLICABLE)
WEIGHTED STYLE
BLOWER & MOTOR MOUNTING BASE
PRESSURE RELIEF VALVE (RUNKLE STYLE)
DISCHARGE SILENCER (HORIZONTAL MOUNTING)
PRESSURE SWITCH (IF APPLICABLE)
SHOULD BE LOCATED ON VIBRATION FREE LOCATION & CONNECTED WITH TUBING TO PRESSURE SOURCE
PRESSURE GAUGE
SUPPORT (BY PURCHASER)
CHECK VALVE (IF APPLICABLE)
AIR PIPE (BY PURCHASER)
¼" ID TUBING & FITTINGS (BY PURCHASER)

NOTES:
AIR PIPING MUST BE INDEPENDENTLY SUPPORTED FROM BLOWER HOUSING FAILURE TO SUPPORT AIR PIPING CAN CAUSE UNDUE STRESS ON BLOWER CAUSING INTERNAL MISALIGNMENT & PREMATURE WEAR.

2. THIS DWG IS FOR EQUIPMENT ORIENTATION ONLY. SEE BLOWER B/M OF CONTRACT FOR ITEMIZED BREAKDOWN OF EQUIPMENT BEING FURNISHED BY FULLER CO.

SUBJECT: TOPIC: 17

PREVENTIVE MAINTENANCE PREVENTIVE MAINTENANCE

OBJECTIVES:

The trainee will be able to:

1) Define the various types of maintenance performed within Preventive Maintenance Program.

2) List the items required to establish a Preventive Maintenance Program.

3) List the priorities for equipment grouping.

4) Set up a record system.

GENERAL

A simplified Preventive Maintenance Program (PM) may be of several different forms ranging from a wall chart on which assigned PM tasks are checked as they are completed, to a more sophisticated system incorporating the use of a small computer. The simplicity and effectiveness of a PM program is not as dependent on the amount and type of equipment to be maintained as it is on the thought and care with which the program is developed, implemented and maintained.

The premise behind all PM programs, regardless of their complexity, is that equipment will operate reliably if scheduled preventive maintenance is performed, unscheduled shutdowns will be reduced. Equipment will be out of service or off-line only at scheduled intervals. To develop and plan a PM program, keep it simple and functional. Do not complicate it or its elements so much that it loses its value. General improvement in equipment reliability will indicate if a PM program is working. An effective program requires some organizational planning and record keeping. To plan a PM program, consider the basic elements. One of the most basic requirements of a program is a complete set of manufacturers manuals for the equipment in the facility. Because these manuals are important, they should be filed in an orderly, easily referenced manner, in a secure location. PM activities may be assigned to the Preventive Maintenance staff or to operations personnel. If daily activities are assigned to operations personnel, a check-off list, providing room for comments and employee signatures should be used. The advantages of assigning daily PM activities to operations personnel is that they are around the equipment frequently and would note any unusual variations in the operation of a unit. Operations staff play an important role in the success of any PM program.

Many PM tasks do not require that a maintenance person perform them, but this does not detract from their importance to the successful operation of a facility.

A PM program provides an effective way to improve equipment reliability and to obtain the information needed to determine which manufacturers equipment is least expensive to operate.

It is essential, however, that the program be well planned and properly implemented, with sufficient feedback and review to keep it on track.

The final step in developing the system is to establish the PM tasks for specific equipment covered. After determining the necessary tasks and the required cycle time (monthly, quarterly, semi-annually, annually), section personnel could estimate the time required to perform the tasks on each piece of equipment.

1. Definition of the types of maintenance for Accounting purposes:

Breakdown Maintenance

Are there any good points?

There should be no good points, as this type of maintenance has no place in a good maintenance program.

What are the bad points?

a) High cost of repairs

b) Shutdown time is high

c) Labour costs are high

d) Interruption of process at a critical time

Routine Maintenance

This type of maintenance is always included in a good maintenance program because it covers the following:

a) A daily check of equipment such as a bearing temperature by touch

b) Oil levels

c) Top up oil levels

e) Grease equipment if required

Maintenance Frequency

An inspection of equipment at regular intervals using a coloured card.

Be able to plan your maintenance work load.

Prevent costly breakdowns.

2. Knowledge of the Process Equipment.

   The most significant preventive maintenance tasks require the shutdown or interruption of the equipment being inspected. In so doing will it have an effect on the process? The superintendent should be informed as to how long it would be out of service. A very important factor to remember is to keep all the mechanical and electrical drawings up-to-date. The flow diagrams should also be brought up-to-date.

3. Data cards for parts and project information. For easy reference to all equipment information. Terminology definitions are as follows:

MOD.	**SER.**	**DIA.**	**V.**	**AMP.**	**c/s or Hz**
Model	Serial	Diameter	Volts	Amperes	Hertz

R.P.M.	**PH.**	**H.P.**	**G.P.M.**	**C.F.M.**
Revolutions per Minute	Phase	Horse Power	Gallons per Minute	Cubic Feet per Minute

4. Frequency maintenance routines to ensure consistent inspection procedures.

That all equipment is inspected at regular intervals.

Prevent any major equipment breakdowns.

Coloured cards:

White - weekly

Blue - monthly
Set up from manufacturers (processes)

Green - 3 months

Orange - 6 months

Yellow - 12 months

Because wastewater treatment facilities are combinations of physical, chemical and biological processes, and loadings on process maintenance activities be well co-ordinated with plant operations. It is also just as important that the maintenance personnel be fully aware of how the process works and understand what effect untimely maintenance activities will have upon the process.

The value of eliminating or nearly eliminating breakdown maintenance in the treatment plant can be easily appreciated. If treatment is to be continuous, the treatment plant cannot let its major pieces of machinery run until they fail, and then effect repairs, unless it is supplied with an over-abundance of standby equipment.

Preventive maintenance, which by definition is scheduled and planned maintenance, is an alternative to an over-abundance of standby equipment. It has never been economically feasible to provide 100% standby equipment required on peak flow days. By providing for the repair and servicing of equipment during periods of the year when service to the public is not critical, scheduled maintenance can provide a reasonable assurance that breakdowns will not occur when the equipment is needed.

To bring breakdown maintenance to a reasonable level, it is necessary to give the maintenance program the same careful attention that is given to plant operations.

The basic features of a sound maintenance policy are:

1. Responsibility for maintenance must be clearly defined and in the hands of qualified, competent personnel.
2. A thorough knowledge of the equipment.
3. Proper tools, spare parts, test instruments and shop facilities for performing maintenance.
4. Preventive action must be planned and scheduled.
5. An adequate system of written records and reports must be used to permit control over the program.
6. Good supervision.

Because the need for maintenance is something that usually does not show, it is important to make a clear and definite assignment of responsibility. When maintenance is everybody's business, equipment breakdown becomes nobody's fault. An assignment of responsibility for maintenance should be delegated to personnel that have the capacity, aptitude and are allowed time for the function.

The fact that the maintenance personnel must be qualified and experienced in maintenance work cannot be over emphasized. Knowledge of such important maintenance factors as what constitutes excessive vibration, when bearings should be replaced, when welding constitutes a safe repair, what a loose fit is, and similar points, are only acquired by experience, training and education.

Preventive maintenance management problems will require less decision making if priorities of maintenance are pre planned. To maintain capital equipment will usually produce one of two results: either the maintenance cost will be high, or the equipment will be poorly maintained causing frequent breakdowns.

For this reason, equipment can be divided into three priority groupings and maintenance dollars spent to best advantage.

The groupings are as follows:

Priority 1 - Equipment that is vital to the process and without such equipment the process cannot achieve desired results.

Priority 2 - Equipment that is considered standby equipment, but may be as reliable as Priority 1 equipment over shorter periods of time.

Priority 3 - Equipment that is not vital to the process, which serves a function other than the process. Wastewater treatment facilities are usually designed for future flow increases, and therefore, the number of pieces of equipment in Priority 1 should be relative to the current flow loadings.

Equipment knowledge can be obtained from instruction manuals issued by the manufacturers of the equipment. These manuals are a result of many years accumulated experience on the part of the manufacturer. Experience is gathered from equipment applications, tests (both field and operational) and operating conditions.

Properly sized shop tools are needed for efficient maintenance procedures. These tools should be of the best quality available. Special tools for certain units should be acquired as needed. The tools a mechanic uses and their condition is a good indicator of the calibre of the mechanic. Repair parts should be stocked according to the number of machines or equipment installed and the expected life of the equipment parts. (The manufacturer's suggestions are often helpful).

Consideration should be made when purchasing spare parts as to the priority allotted the individual piece of equipment. Priority rating is established by asking the following questions:

1. Is this piece of equipment vital to the process?
2 What maximum down-time can be tolerated if the equipment fails at the worst possible time?
3. Is standby equipment readily available?
4. Can other equipment fail due to load changes caused by a failure of this piece of equipment?

Preventive maintenance is successfully practiced when it is properly planned and performed according to a prearranged schedule. It requires the same attention and importance as the transfer of sludge, digester temperature, or the cleaning of bar screens. Unless preventive maintenance is routinely performed and regarded as a plant operation, as are actual plant operations, breakdown maintenance cannot be avoided.

The planning of a preventive maintenance program, therefore, requires that equipment items be studied to determine the maintenance operations required and the frequency of these operations. This is determined from an analysis of service conditions of the equipment, from your operating experiences, from others' operating experiences and from experiences and from equipment manufacturer's recommendations. Each equipment item or unit must be studied individually, for even similar pieces o' equipment may have different care requirements because of location or service.

To carry out and administer a maintenance program effectively requires a minimum of records and forms, yet such records are very important to successful maintenance for the following reasons:

1. Provide data for accurate budget forecasting.
2. Prove compliance with manufacturer's instructions.
3. Validate warranty conditions.
4. Evaluate equipment performance.
5. Highlight recurring faults.
6. Establish maintenance frequency.
7. Comparison of existing equipment.
8. Evaluate the maintenance program.

In summary, continuity of service and performance is of prime importance in a wastewater treatment facility. Good maintenance is good management and operation. A preventive maintenance program is scheduled routine and frequency maintenance that will cut breakdowns and unplanned at peak efficiency. No treatment facility is too small to install or enough to be able to afford not to take care of its equipment.

Summary

The general items required to establish a preventive maintenance program should include:

1. Routine maintenance activities.
2. Frequency maintenance activities.
3. Complete diagrams and "as built" prints of the facility.
4. Process equipment data cards and operating manuals.
5. A record keeping and reporting system.

A good preventive maintenance program will:

1. Cut down on costly equipment breakdowns.
2. Reduce the need for 100T standby equipment.
3. Prevent the loss of vital process stages that would reduce effluent quality.

4. Provide effective and efficient operation.

5. Utilize manpower more effectively.

6. Increase the cost of stocking replacement parts.

7. Comply with manufacturer's warranties.

8. Decrease equipment "down-time".

9. Equalize the maintenance workload.

10. Require competent and qualified personnel.

GLOSSARY

Brake Horsepower

The actual horsepower of the drive required to produce a given water H.P.

Dynamic Discharge Head

Is the vertical distance in feet between the pump centre line and point of free discharge or to the level of the free surface of the discharge water, plus velocity head and friction losses in the discharge piping

Dynamic Suction Head

Includes static suction head minus the sum of friction head loss and velocity head.

Dynamic Suction Lift

Includes the sum of static suction lift, friction head loss, and velocity head.

Efficiency of a Pump

The ratio of the output (water horsepower) to the input (brake horsepower).

Energy

The capacity to perform work. It is expressed in units that are the product of weight and length. The unit of energy is the product of a unit weight by a unit length, i.e. the foot-pound, which is the energy required to move one pound one foot.

Energy exists in two forms:

1. Potential Energy - that due to position or elevation.
2. Kinetic Energy - that due to its velocity of motion.

Friction Head

The pressure expressed in lbs./sq.in. or feet of liquid needed to overcome the resistance to the flow in the pipe and fittings.

Liquid Horsepower (water horsepower)

This is the theoretical horsepower required to pump water at a given rate against a given head, and is obtained from the formula:

$$\text{water h.p.} = \frac{\text{U.S.G.P.M.} \times \text{head in ft.} \times \text{spec. grav.}}{3960}$$

$$\text{or} \quad = \frac{\text{I.G.P.M.} \times \text{head in ft.} \times \text{spec. grav.}}{3300}$$

The constant 3960 is derived by dividing the number of foot pounds of one H.P. (33,000) by the weight of one U.S. gallon of water (8.33 lbs.).

Power

The rate of performing work, and is expressed in horsepower, 550 ft-lb/sec, or kilowatts, 737 ft-lb/sec.

Rating (or rated capacity)

This describes the condition for which the pump is designed. A pump is said to be rated at a given capacity for a given head. This condition generally occurs at the point of maximum efficiency.

Specific Gravity

Direct rating of any liquid's weight to the weight of water at 62°F. Water 62°F weighs 8.33# per gallon and is designated 1.0 sp. gr.

Specific Speed

A speed or velocity of revolution, expressed in revolutions per minute at which a pump of suitable size and proportion, would develop one-foot of head if powered by one horsepower motor used for pump design.

Static Discharge Head

The vertical elevation from the centre line of the pump to the point of free discharge.

Suction Head

Exists when the source of supply is above the centre line of the pump.

Suction Lift

Exists when the source of supply is below the centre line of the pump.

Static Suction Head

The vertical distance from the centre line of the pump up to the free level of the liquid source.

Static Suction Lift

The vertical distance from the centre line of the pump down to the free level of the liquid source.

Total Dynamic Head

The vertical distance in feet from the free liquid level to be pumped, and point of free discharge, or to the level of the free surface of the discharge, plus velocity head and friction losses in the suction and discharge piping.

Velocity Head

The head needed to accelerate the liquid. Knowing the velocity of the liquid, the velocity head loss can be calculated by a simple formula: $\underline{\text{Head} = V^2/2g}$ in which g is acceleration due to gravity or $32.16\ \text{ft/sec.}^2$.

Viscosity

Property of a liquid that resists any force tending to produce flow. It is the evidence of cohesion between the particles of a fluid which causes a liquid to offer resistance analogous to friction. A change in the temperature may change the viscosity depending upon the liquid. Pipe friction loss increases as viscosity increases.

GLOSSARY OF PUMP CONTROL TERMS

ACCURACY	is the degree of correctness or freedom from error.
ALTERNATOR	is a relay or switch used to alternate the starting of say, two pumps which automatically cycle on and off. Alternator also means AC generator.
AMPLIFIER	is a signal gain device whose output is proportional to its input.
ANNUNCIATOR	is a device with one or more lamps and sometimes a common buzzer which operate on a failure or particular condition.
BREAKER or **CIRCUIT BREAKER**	is an automatic electro-mechanical device which can trip a predetermined current.
CONTACTS	are conducting parts which can complete or interrupt a circuit.
Auxiliary	are in addition to the main circuit contacts.
Relay	are referred to as normally open (NO) or normally closed (NC) which is their state with the relay de-energised.
CONTRACTOR	is the part of a magnetic starter which operates the contacts by a magnet coil in an iron yoke. In other words, it is a large relay handling the supply to the motor.
CONTROL(S)	is the means of governing the performance of a machine or system.
Automatic	is obtained by using sensors, auxiliary relays and safety devices.
Auxiliary	connect the sensing signals to the pump starters such as relays, time delay relays, timers, interlock devices.

Bubbler	is a level control using low pressure air on diaphragm sensors using a small compressor, air regulators and pressure relays.
Circuits	are the wires and devices which control the starter as distinct from the "power circuit" which supplies the pump motor.
Loop or Feedback	is obtained by comparing the pump performance with a reference and any difference is arranged to control the pump.
Manuals	are switches, disconnects, push buttons, "HOA" selectors, manual starters, manual speed rheostat etc.
Program	arranges for one or more machines etc. to perform a series of operations.
Remote	can operate the pump from a remote point.
Sensing	are required for automatic operation of the pump.
System	is all the control equipment required to operate the pump automatically.
Transformer	usually transforms the voltage from the power supply (600V) down to the control supply (120V). This is distinct from the power, distribution or lighting transformers.
CONTROLLER	see starter.
DISCONNECT	see switch.
DUTY	refers to the type of service such as "Continuous"; "Intermittent"; "Lead or Lag"; "Duty" or "Stand-by", etc.
Duty Cycle	refers to the pump duty e.g. a 10 minute duty cycle could be 4 minutes ON plus 6 minutes OFF. (This would be a 40% duty cycle)
Duty Pump	refers to the in service pump as compared with the stand-by or lag pump.

ELECTRO-MECHANICAL	is where the components are electrical (contacts & coils) together with moving parts. This is distinct from solid state.
ELECTRO-PNEUMATIC	uses electrical and pneumatic components.
GROUND DETECTOR	is used with ungrounded systems and is an indicator with 3 lamps or neons to detect if one phase of the 3 phase supply is grounded.
INTERLOCK	permits or prevents the operation of a switch, breakers, relay, etc. in a predetermined sequence so as to govern succeeding operations.
LOGIC DIAGRAM	by symbols illustrates the operating functions of a control system.
MOTOR CONTROL CENTRE (MCC)	a group of motor starters arranged in one switchboard for ease of control. Other devices are often included in the switchboard.
OVERLOAD TRIPS and OVERLOAD HEATERS	see overload relays.
PANELBOARD	is a group of moulded case circuit breakers in metal enclosure to control branch circuits.
RATING	is the duty or service of a motor such as "continuous" or "short time" or "intermittent". Rating is also the HP(KW), voltage, current, etc.
RELAY(S)	is a device which when it operates will control the performance of another device.
Auxiliary	is usually an intermediate relay to provide a sequence or multiple function.
Control	is similar to an auxiliary relay and always handles low energy-level control signals.

Latching	is where the contacts are held mechanically on the first electrical impulse and unlatch and release the armature on the second impulse. In certain relays the latching alternates from the energised to de-energised position as in the case of an alternator relay.
Overload or Overcurrent	is a thermal relay usually in the starter to protect the motor against overload. Reset is usually by manual push-button but automatic reset is provided in special cases.
Reset	restores the control or alarms to their normal operating condition.
Temperature	opens a contact when over-temperature condition is sensed (e.g. winding or bearing).
Time Delay	is where the contacts close, or open, after an intentional time interval which is usually adjustable.
Transfer	is usually an auxiliary relay which operates a transfer switch. In small ratings it could be the transfer switch itself.
Under- or Over-Voltage	operates at a predetermined under- or over-voltage voltage to trip the starter and protect the motor.
RESET	is a push-button or relay to restore the overload trip, relay or alarm to the normal operating condition.
SENSOR(S)	
Level	is to operate a signal according to the liquid level such as a float switch, capacitance or conductance probes, "bubbler", sonar detectors pressure or vacuum switches.
Pressure	is to operate a signal according to the liquid or air pressure such as diaphragm or bellows switches.
Temperature	is to operate a signal, contacts or trip bar according to the temperature of the motor or transformer etc. These can be "inherent" such as in the motor winding.

STARTER(S)	is for accelerating a motor from rest to normal speed.
Automatic	automatically controls the acceleration of the motor.
Autotransformer	provides a reduced voltage for starting by the autotransformer. Transition from reduced to full voltage can be by "open" or the preferred "closed" method. This starter is sometimes called compensator or autostarter.
Combination	is a single package containing an interlocked disconnect switch with fuses or preferably a circuit breaker, contactor and overload relay(s)
Full Voltage	is where the full supply voltage is applied directly to the motor through the starter contacts.
Magnetic	is equipped with a contactor which can be operated locally or remotely by push-buttons or automatically.
Manual	is where the contacts in the motor circuit are closed and opened manually and will open (trip) automatically on overload.
Part Winding	is where two contactors are closed in sequence to energise the two halves of the motor winding in two steps.
Primary Resistor	(or Reactor) is where the voltage is reduced for for starting by a primary resistor (or reactor) before the full voltage is applied by a second contactor.
Reduced Voltage	is a general term as distinct from full voltage and includes primary resistor, reactor, auto transformer, part winding and star delta starters.
Reversing	has two interlocked contactor one for the motor forward direction and one for reverse direction.

Star Delta	has a start contactor which supplies the motor windings in a star arrangement. This contactor is interlocked with the sun contactor which subsequently supplies the motor windings in a delta arrangement.
Two Speed	has two contactors, one for the motor low speed operation and one for high speed.
Variable Speed	in a much more complex starter and includes primary and secondary contactors, step speed resistors or special solid state voltage control.
Wound Rotor	has a primary contactor together with a number of secondary contactors and resistors.

STARTER ENCLOSURES

General Purpose	has a metal case to protect against accidental contact with live parts and is for indoor service (Type 1).
Dust Tight	is for indoor service and excludes dust or fibres from entering the starter (Type 5).
Drip Tight	is for indoor service, similar to general purpose type but with drip shields (Type 2).
Explosion Proof	is a very special enclosure to suite the type of hazardous location.
Weather Resistant Protection	is for outdoor service and gives protection against rain or sleet (Type 3).
Water Tight	gives protection against water from a hose (Type 4).
Steady State	is the condition when no transients or "fluctuations" are occurring.
Solid State	is where the components are not moving or electron tubes i.e. transistors, SCR's , etc.

SWITCH(ES)

Cam Operated	is where the contacts are operated by a drum, disc or cam.
Disconnect	is a manual switch in the supply to the motor or starter to isolate them for service.
Drum	is usually associated with a variable speed wound rotor motor and where the cams and contacts are arranged in the form of a drum.
Float	is used as a sensor to detect when a water level has been reached.
H-O-A	has 3 positions "Hand" "Off" "Automatic".
Isolating	see Disconnect.
Limit	is used to control the extent of travel e.g. on a motorized valve.
Manual	as distinct from a solenoid, motor operated or static switch.
Pressure	is a sensor which operates at a preset air or water pressure.
Selector	is usually manually operated to select a function e.g. H-O-A, Ideal remote.
Service Entrance	is usually the first switch within the building which controls the electric supply.
Speed	(Overspeed) is a sensor which operates on the motor speed.
Transfer	transfers the load from one sources (Hydro) to an alternative source (local generating set).

SYSTEM (POWER SUPPLY)	
Grounded	is where a neutral point in the supply is directly connected to ground rods at the local transformer station.
Insulated	is where the supply is not directly grounded and is left "floating" monitored by ground detectors.
Low Potential	is where the supply is between 31 and 700 volts.
High Potential	is where the supply is over 700 volts.
TIMERS	are used to control a sequence of operations e.g. backwash filters. Sometimes time delay relays are called timers.
TRANSIENT	is a temporary abrupt departure from steady state conditions.
TRANSITION	is the term used with reduced voltage starters covering the transfer from the reduced voltage to the full voltage by interrupting the supply "open transition" or not interrupting the supply "closed transition".
TRIPS (Thermal magnetic)	is usually associated with circuit breakers to provide a time delay and instantaneous combination of circuit protection, depending on the magnitude of an overcurrent condition.